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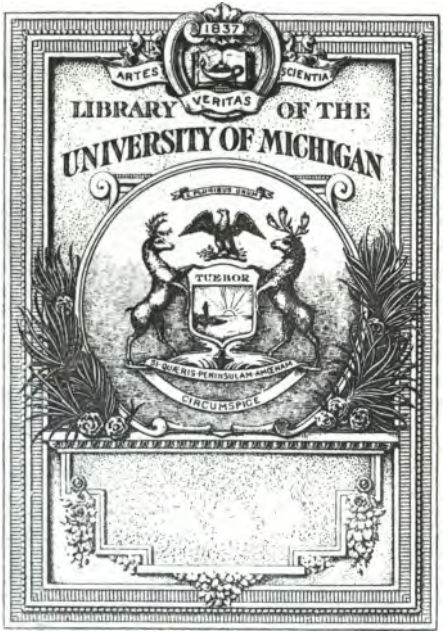

THE GIFT OF
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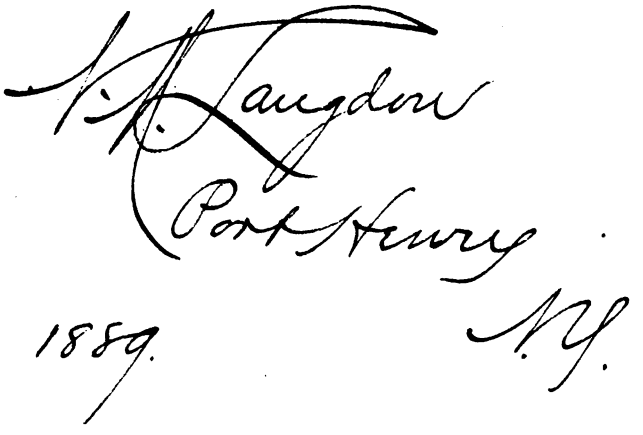

SLIDE VALVE GEARS.

A

NEW GRAPHICAL METHOD

FOR

ANALYZING THE ACTION OF SLIDE VALVES,

MOVED BY

ECCENTRICS, LINK-MOTIONS AND CUT-OFF GEARS,

Offering easy means for properly designing Valves and Valve-Gears, and for establishing the comparative merits of their various constructions,

BY

HUGO BILGRAM, M. E.

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PREFACE.

The most successful method of studying and analyzing the action of the slide valve, in its various modes of application, is that which is based on the use of graphical diagrams, among which the one discovered by Zeuner,* is pretty generally considered the best.

This diagram shows at a glance what, by the aid of working models, can be seen only by carefully watching the movement of the slide valve in relation to that of the piston; it shows how an alteration of any of the essential members of the valve gear affects the working of the valve; it is admirably adapted to draw a comparison between different valve gears and link motions on their intrinsical as well as their relative merits; it also furnishes the means, by a reversed application, of finding the proper proportions of valves and the valve-moving devices when a given effect is desired, and how any valve gear can be proportioned to the best advantage. In short, it meets all the principal requirements of the case. Only one feature is in some respects adverse to a general use of this graphical method; in some instances its application requires algebraical calculations, or else an intricate geometrical procedure, and for this reason it can be used only by those engineers who have enjoyed a theoretical education.

In his efforts to fully open this field to every intelligent mechanic, the author found that a modification of the Zeuner

*Schiebersteuerungen, by Prof. Dr. Gustav Zeuner

Diagram renders the subject in all its bearings purely graphical, and no knowledge of mathematics beyond the most elementary propositions of geometry is required to understand the demonstrations as well as the applications of this new diagram. Nor is this the only advantage gained over the original form; for by this modification the diagram profited in several other respects, especially in regard to clearness, and will therefore give better satisfaction, even to engineers familiar with mathematics.

The following discussions will be as brief as is consistent with a thorough and fundamental treatment; and the leading propositions will be amply illustrated.

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INTRODUCTION.

Proper Distribution of Steam.

The steam engine is a mechanical contrivance for converting the power, developed by the evaporation of water, into such a form that it can be utilized for driving machinery or for other industrial purposes. After its generation in the boiler, the steam is caused to exert its pressure alternately upon both sides of the piston, imparting a reciprocating motion, which is converted into a rotary one by a connecting-rod and crank. The valves admitting the steam alternately to both ends of the cylinder, and opening an exit for the steam after the performance of its duty, are named "distribution-valves" and the mechanism operating them is termed "valve-gear."

We shall now consider how the admission of steam should be regulated in order that the greatest amount of useful effect will be realized; this will enable us to judge how near the valve-gears to be considered hereafter approximate to perfection.

I. *Expansion.* To the engineer, the most valuable property of steam is its elasticity, which can most effectively be utilized by admitting steam to the cylinder for a portion of the stroke only. The production of work per stroke will then comprise two stages: (1) The direct action of steam, for the time of admission, when the steam exerts its initial pressure upon the piston, and (2) The action of the expanding steam; in which period the pressure is gradually diminishing.

For economy of steam it is most advantageous to expand

until the terminal pressure upon the piston equals the average frictional resistance of the engine. In practice, however, the expansion is not always carried thus far, as an engine accomplishing this result generally demands a higher price than one with a single slide-valve, and purchasers of engines frequently prefer to save in the first cost.

When the load on the engine is subject to variation, the admission of steam must be regulated accordingly. The effective steam pressure upon the piston is usually more or less reduced by throttling the admitted steam by a valve that is operated by the governor. A more economical plan, however, is to vary the degree of expansion by cutting the steam-off at an earlier or later stage of the stroke, if the valve-gear admits of such a variation.

II. *Compression.* That end of the cylinder towards which the piston is moving must be in communication with the exhaust passage. In condensing engines the exhaust passage is in communication with the condenser, where a partial vacuum is produced by the condensation of the exhaust steam. Other engines exhaust into the air, and have therefore a back pressure equal to, or slightly above, that of the atmosphere.

It is of advantage to interrupt the exhaust a short time before the termination of the stroke, and to compress the thus insulated exhaust steam by the piston. At the first glance this statement may appear absurd; but it is nevertheless consistent with both theory and practical experience, for the following reasons:

The "clearance,"—which is the space left at the end of the stroke, between the piston and the end of the cylinder, together with the space of the steam passage between the valve and the cylinder,—is alternately brought in communication with the steam chest and with the exhaust passage, and is thus filled with steam and emptied again. The direct

action of this steam is evidently lost, and only part of the expansive power of the same is made available. This continual loss of power can, however, be moderated by compressing into the clearance part of the exhaust steam, raising its pressure to that in the boiler. When thereafter the valve opens the port for admission, no steam is required to fill the clearance, and the full power of all the admitted steam will be turned into useful work. If now the steam is cut off so early in the stroke that the terminal pressure will be equal to that in the exhaust passage, the re-expansion of the previously compressed steam will augment the production of work just so much as to restore the work expended in compression. This shows that under the said conditions the clearance will cause no loss of steam. It must, however, be remembered that in the first place we took no account of friction. The space through which the piston travels during compression, is entirely lost for the production of work; therefore it is evident that an engine with much clearance and corresponding compression, though theoretically equal to one with but little clearance and compression, is practically inferior, for since the travel of the piston must be greater it will involve greater loss from friction. In the second place, we assumed the expansion carried to a degree higher than that generally used. It is, therefore, advisable to compress to a less degree, especially if not much expansion is used; for the work expended for compression will be but partially restored by the following re-expansion.

A second reason for prematurely closing the exhaust is due to the reaction of the compressed exhaust steam upon the piston,—commonly called "cushioning,"—which assists in the reversal of the motion of the piston and other reciprocating parts of the engine and lessens the wear in bearings, and also the thumping caused by lost motion which is usually present to some extent.

III. *Lead.* Experience has taught that both the admission and the exhaust of steam should be commenced a short time before the crank passes the centre. Either port will thereby be opened to some extent at the beginning of the stroke. The width of the opening at that time is termed "lead," while the angle through which the crank passes from the moment of the opening of the port until it reaches the centre is the "lead angle."

One reason is obvious. The steam does not fill the clearance, nor does the exhaust steam reduce its pressure to that of the exhaust passage, the very instant the port for either admission or exhaust is opened; hence the advantage of a premature opening on either side.

As a general rule the lead on the exhaust side is greater than that on the steam side, as the former has the function of releasing the pressure of a volume of steam filling the entire cylinder, while the latter need only fill the clearance with live steam. The opening of the *exhaust* is mostly effected while the piston is still an appreciable distance from the end of its stroke, and is more commonly called "release," while the *admission* mostly begins when the piston has practically arrived at the end of its travel.

The most advantageous amount of lead on the exhaust side depends mainly on the terminal pressure of the expanded steam; and should, therefore, be greater on engines with late cut-off. High speed engines need more lead on both sides than slow running ones, to allow the proper time for the functions of lead. Excessive lead on either side always results in waste of power.

Lead is also a very convenient means for supplementing the *cushioning* if compression is insufficient for this purpose; it can be adjusted until the engine ceases to thump. High speed engines and beam engines need more lead on this account, than others, because of the greater momentum to be overcome.

PART I.

THE COMMON SLIDE VALVE.

The Common Slide Valve.

The usual form of the common slide valve is shown in Fig. 1, where S and S' represent the steam passages of the cylinder,

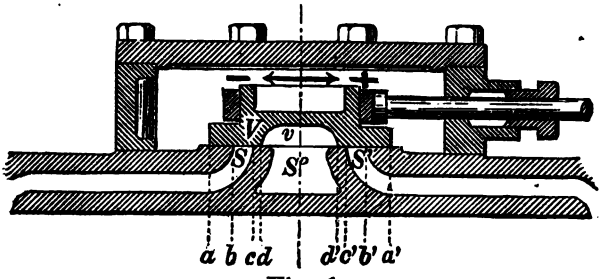

Fig. 1.

and S^0 the exhaust passage. V is the valve, the cavity v of which effects alternately a connection of the steam ports with the exhaust port. The valve is encased in the steam chest, into which the live steam is admitted. It is moved by the valve stem passing through a stuffing box of the steam chest.

When the valve is in its neutral position, as shown, its face overlaps the steam ports on both sides. The distance $a\ b$ is called the outside or steam side lap, and $c\ d$ is the inside or exhaust side lap.

In the following considerations we shall term "positive" any movement of the valve from its neutral position towards the valve stem, in contradistinction from a movement in the opposite or "negative" direction. A positive movement of the valve connects the port S' with the exhaust passage S^0 through the cavity v, and opens the port S for admission of steam from the steam chest. The width of the opening of either port depends on the distance of the valve from

its neutral position, for the time being. This distance, or the "movement" of the valve, must exceed the inside lap before he port S will be opened for exhaust; and it must be greater than the outside lap, to effect an opening of the port S for admission of live steam. For a given movement of the valve we find the opening of the steam ports for either exhaust or

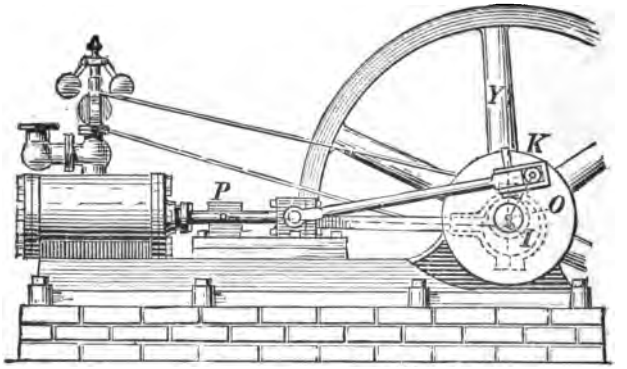

Fig. 2.

admission by deducting from this movement the inside or outside lap respectively. When this difference is equal to or greater than the width of the port, the latter is fully opened. The effect produced by a motion of the valve in the opposite (negative) direction is similar, but reversed as regards the passages S and S'.

The valve is moved by means of the eccentric I, (Fig. 2), which can properly be considered as a small crank, the pin of which is enlarged so much as to assume the form of a disc that allows the shaft O to pass through it. The eccentric can therefore be represented by the line OI, that connects the centre of the shaft, O, with the centre of the eccentric disc, I, and which is supposed to act like a crank.

The Valve Diagram.

We shall first consider horizontal engines of the usual construction, in which both the connecting rod and the eccentric rod, are operated in parallel directions.

The eccentric of such engines is keyed to the crank shaft so that the centre line *OI* (Fig. 2) of the eccentric forms an obtuse angle with the centre line *OK* of the crank. As the crank shaft rotates, the end *P* of the eccentrie rod, being guided in a right line by the valve stem, receives a reciprocating motion while the rod itself is caused to oscillate. The reciprocating motion of the end *P* is transmitted by means of the

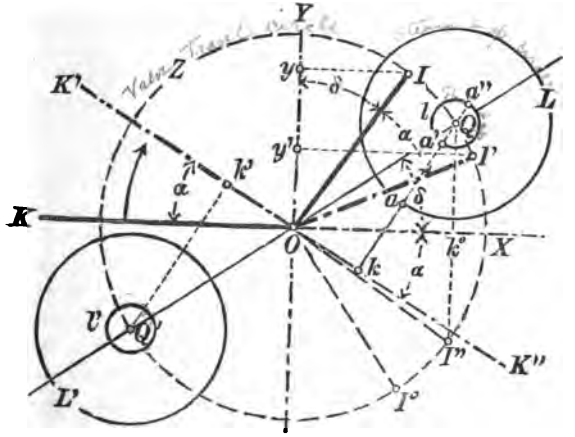

Fig. 3.

valve stem to the slide valve. The oscillating movement introduces a very intricate factor, since it affects the horizontal extension of the eccentric rod. We shall, however, at present

leave this factor altogether out of account, and shall examine it later.

It is necessary to locate the centre of vibration of the end P of the eccentric rod on account of its corresponding relation to the neutral position of the valve. The vibration of this point being a direct result of the circular movement of the other end, I, of the eccentric rod, it is evident that the centres of both movements must correspond. Therefore, if the end I of the rod IP be transposed to the centre of its motion, (which is the centre O of the crank shaft) its other end P will occupy the centre of its vibration. It will now be understood that the movement of the pin P depends on the horizontal movement of the centre, I, of the eccentric; and can therefore be measured by the horizontal distance of the point I from the vertical centre line OY.

When the crank is on its centre OK (Fig. 3) and I is the corresponding position of the eccentric, the distance Iy will represent the valve movement.

As the crank moves from K to K' through the angle a, the eccentric will move through the same angle to I', and $I'y'$ will now be the distance of the valve from its neutral position.

Let us now mark the point Q in the circle Z representing the orbit of the eccentric; making the angle XOQ equal to the angle IOY (both of which are marked δ in Fig. 3), and draw the line Qk perpendicular to the extension of the line OK' of the crank. The triangles QOk and $I'Oy'$ will then be equal, since the angles QOk and QkO are equal respectively to the angles $I'Oy'$ and $I'y'O$, and the side OQ equals the side OI'; hence the line Qk equals $I'y'$ and therefore is equal to the movement of the valve for the crank angle a. The same relation can be shown, in the same manner, for any other crank angle, and the interesting fact is thereby demonstrated that, *for any angle of the crank, the distance of the*

point Q from the centre line of the crank is equal to the corresponding movement of the slide valve.

To find the openings of the ports for either admission or exhaust, the inside or outside lap is to be deducted from this movement. This subtraction can be graphically performed by sweeping two circles, L and l, from Q with radii equal to the outside and inside laps, as shown in Fig. 8. For the considered position, K', of the crank, the movement of the valve was found to equal the distance $Q\,k$, and hence the opening of the steam port S (Fig. 1) for admission of steam is equal to ka, and that of the port S' for exhaust equal to ka'; these distances being equal to the movement Qk, minus the respective laps. The port openings are therefore represented by the shortest distances between the centre line of the crank and the lap circles L or l, respectively.

Had the point Q', instead of Q, been selected, the result would have been the same, since $Q\,k$ equals $Q'k'$.

To prevent any misunderstanding it may be here stated that Fig. 8 now represents two distinct diagrams, one showing the position of the eccentric, I, in relation to the crank, OK, when on its centre, and another in which the point Q figures and which will be called the "valve diagram." The latter can be obtained from the former by making the angle XOQ = angle YOI, or also, by rotating the first diagram through a right angle, when the eccentric will arrive at I," and subsequently inverting it. This last mode of converting one diagram into the other, can very conveniently be performed by means of a piece of tracing paper, viz., by tracing the diagram of crank and eccentric, KOI, turning this tracing over and setting the crank-line, OK, in a vertical position, pointing downwards. The position of the eccentric, I, will then locate the point, Q, whose mate, Q', is diametrically opposite.

In order to know the direction of the valve-movement we

must consider that the valve will be in its neutral position when the crank passes through Q or Q', the distance of Q from the crank-line being zero. As the crank moves from Q' through K' to Q, the movement of the valve is positive, and while moving from Q to Q' this movement is negative.

When the engine is required to rotate in an opposite direction, the position of the eccentric will be I°, in relation to the crank, OK; and the valve diagram will be reversed or left-handed.

The valve diagram as explained affords an easy means of studying the influence of form and motion of the valve on the distribution of steam. The most important positions of the crank, as regards the admission and the exhaust of steam, are shown in Fig. 4. When the crank is on its centre, at K,

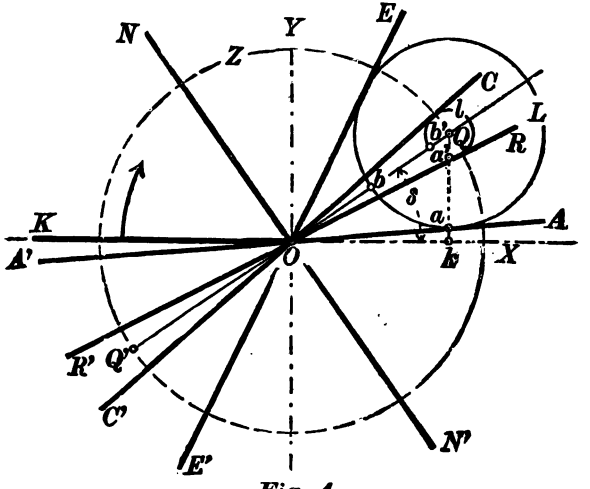

Fig. 4.

the distance *ka* shows the opening of the port for admission, or the lead on the steam side, and *ka'* represents the opening of the port for the exhaust, or the lead on the exhaust side. The valve attains its maximum movement when the crank is at *N*, forming right angles with the line *OQ*; hence the greatest port-opening for admission of steam is equal to *Ob*. This distance may be greater than the width of the steam port, showing that the outside edge of the valve overruns the inside edge of the port, allowing an unobstructed admission of steam. The word "port-opening" will be used independently of the width of the steam ports, and may therefore exceed the latter. As the crank proceeds, the port-opening will be reduced again; and when the crank arrives at *E*, the port, *S*, (Fig. 1) will be closed and the admission of steam cut off; the valve movement being equal to the outside lap. At *O* the valve movement will be equal to the inside lap, hence the port, *S'*, (Fig. 1) will be closed, and compression will commence. *OQ* is the position of the crank when the valve passes its neutral position At *R* the negative movement of the valve equals the inside lap and the port, *S*, is opened for exhaust. At *A* the port, *S'*, will be opened for admission of steam for the following stroke. For the crank-angles *N', E', C', R', A'*, the results will be merely a repetition of what has already been said, with this difference only: that the ports exchange their respective functions. The lead angle for admission is shown in the diagram by the angle *A' OK* or *AOX*, and that for exhaust by the angle *R' OK* or *ROX*.

This diagram may perhaps seem, to a beginner, somewhat complicated, especially as it is necessary for some crank angles to extend the crank-line over the centre, *O*; and further, as it requires some practice to determine which of the two ports is being opened or closed. It can, however, be so drawn that the operation of the valve on each port may be represented separately, as in Fig. 5, relating to the port, *S*.

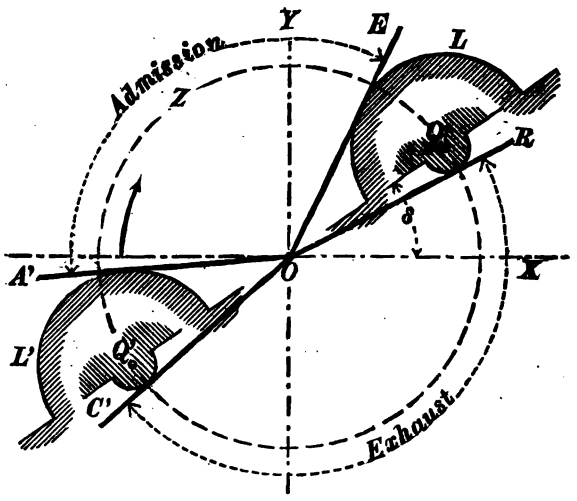

Fig. 5.

(Fig. 1) and in Fig. 6, relating to the port, *S'*. By a comparison with Fig. 4, these diagrams may be readily understood without further comment. These diagrams indicate the port to which they relate to be opened, for either admission or exhaust, as long as the crank-line is outside of the respective cross-lined surface and each of them may again be regarded as combining two diagrams; one for admission, and the other for exhaust.

These two diagrams, Figs. 5 and 6, can readily be combined by drawing the upper half of Fig. 5 and the lower half of Fig. 6 in full lines and the rest in dotted lines. Some of the following diagrams will be executed in that manner; and it is only to be remembered that the crank-angles for cutting-off and for release (relating to the port on the steam side of the

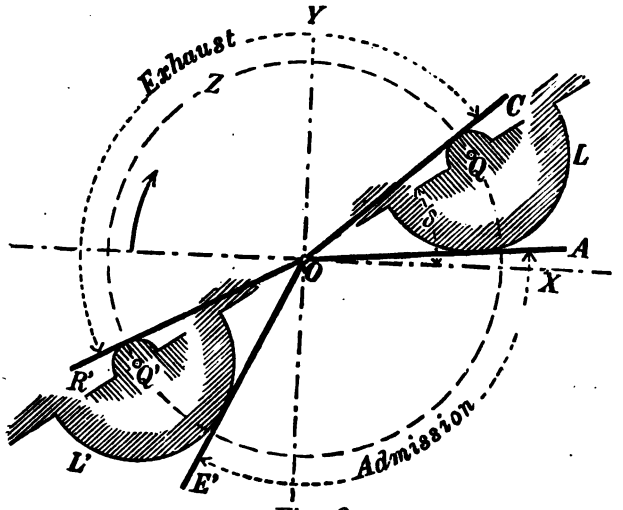

Fig. 6.

piston) will always be tangent to solid circles, while the crank-angles for compression and the beginning of admission (relating to the port on the exhaust side of the piston) will be tangent to the dotted lap circles.

When the valve possesses no inside lap, the inside lap-circle shrinks into the point, Q, and hence Q and Q' are the positions of the crank where the exhaust is closed for one port and at the same time opened for the other port. Sometimes the valve may be constructed as shown in Fig. 7, possessing the converse of an inside lap. The distance, *cd*, is properly called "negative lap." Some engineers call it "inside lead," others, again, "inside clearance," but both these terms are apt to lead to confusion, for "lap" is measured when the valve is in its neutral position; "lead" is measured when

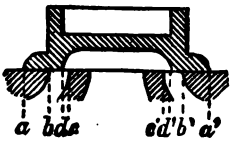

Fig. 7.

the crank is on its centre; and "clearance" is the conventional name for the space left between the steam-valve and the piston at the end of its stroke. To find the opening of the port for exhaust in such a case, the distance, *cd*, must evidently be added to the movement of the valve, and hence the opening is not *ka'* (Fig. 3) but *ka''* and the diagram represented in Fig. 5 becomes like Fig. 8, which shows that in

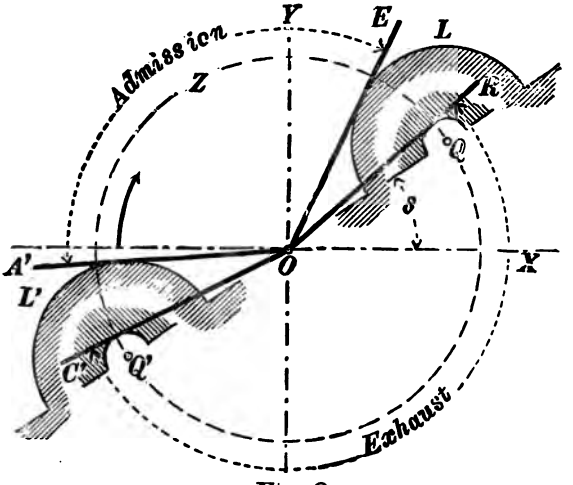

Fig. 8.

this case the exhaust remains opened for more than a semi-revolution of the crank shaft.

It has been stated before that the greatest port-opening of the valve sometimes exceeds the width of the port. The diagram shows without difficulty the angles for which the port remains opened to its full width. We sweep a circle, z, from

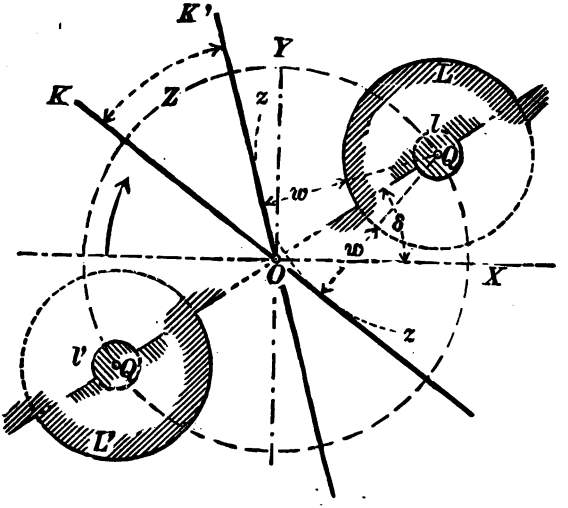

Fig. 9.

Q (Fig. 9), the distance of which from the lap circle, L, is equal to the width, w, of the port. For the two crank angles, OK and OK', which form tangents to this circle, the port-opening will be equal to the width of the port, and between K and K' it will be in excess of the latter.

Or, it may happen that the greatest port-opening for admission is less than the width of the port, which then will

become fully opened for the exhaust only. The greatest port-opening for admission should, however, never be less than 75 per cent. of the width of the port.

The manner of comparing the motion of the crank with that of the piston will be explained later.

The Angle of Advance.

On horizontal engines the centre-line *OI* of the eccentric forms, as already mentioned, an obtuse angle with the centre line *OK* (Fig. 2) of the crank. This relation will be modified when both the connecting rod and the eccentric rod are not operated in parallel directions, as may be the case with beam engines, or where a rocker or some other lever connection is interposed, for instance as shown in Fig. 10. But no matter

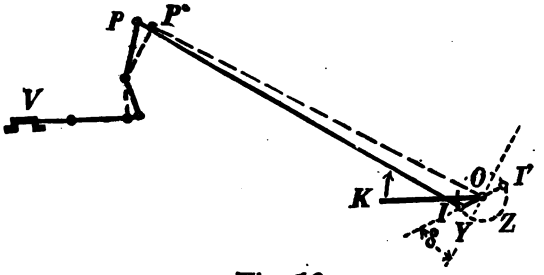

Fig. 10.

whatever the general arrangement of the engine may be, the centre line of the eccentric forms most always approximately right angles with the eccentric rod when the crank is in line with the connecting rod, or "on its centre." It should, however, never be precisely at right angles, but always more or less in advance of this position, in the direction of the rotation of the crank shaft, for else the valve would be in its neutral position at the beginning of the stroke; and since the port

on the steam side is required to be opened, the eccentric must be advanced to make the lap and lead. The angle measuring this advance, namely, the angle formed between the eccentric line OI (Fig. 10) and the line OY drawn at right angles to the mean direction of the eccentric-rod, is called the "angular advance" or "angle of advance." The mean or neutral direction of the eccentric rod may be determined, in the drawing, by transferring the end I of the same into the centre O of the crank-shaft, and is shown, in Fig. 10, by the dotted line OP°. Whenever the end P of the eccentric-rod is guided in an arc, as in Fig. 10, this construction does not give a perfectly accurate result, but it will be sufficiently accurate for practical purposes, as the valve performs its principal operations when near its central position.

The "angular advance" plays an important part in the action of the slide valve and shall be denoted by the letter δ, as has already been done in the former diagrams.

In order to investigate the operation of its valve motion, any type of engines may be compared with a horizontal engine having the same slide valve, with equal travel, and equal angular advance. The valve diagram can therefore be drawn by first sweeping a circle for the path of the eccentric and then carrying in it the angle of advance from the horizontal centre-line in a direction opposite that of the rotation of the crank. From the points thus obtained in the circumference of the circle the lap circles have to be swept to complete the diagram.

It need hardly be mentioned that in cases where the motion of the eccentric-rod is transmitted to the valve by rockers or levers of unequal leverage the radius of the "path of the eccentric" of the diagram should not be made equal to the throw of the eccentric, but equal to one-half the travel of the valve.

In designing engines other than such where the connecting-rod and eccentric-rod are operated in a parallel direction, beginners often find difficulty in drawing the eccentric in its proper position to the crank, the angle of advance being given. The course to be pursued in such cases is as follows. Draw the crank OK (see Fig, 10) on one of its dead centres, and then draw the line OY, at right angles to the mean direction, OP°, of the eccentric-rod and carry the angle of advance, δ, from the last mentioned line, in the direction of the rotation of the crank-shaft. The resulting line determines the angle of the eccentric, but it must be decided yet whether I or I' (see Fig. 10) is the correct position. This can be done by assuming the movement of the crank to proceed, and considering which of the two points I or I' would cause the valve to open that port that will admit the steam on the proper side of the piston.

Width of Ports.

The sectional area of the steam-ports should be made so large as to impede the ingress and egress of the steam as little as possible, and yet, on the other hand, so small as not to unduly increase the amount of clearance. By a series of carefully conducted experiments, it was found that the greatest velocity of the steam in the passages should be from 8,000 to 9,000 feet per minute, in small engines and such of a low pressure and low mean piston speed, and from 9,000 to 10,000 feet in large ones and such of high piston speed.

To compute, from these data, the area of the ports in any one special case, the maximum velocity of the piston must be found, which is equal to that of the crank pin, and hence equal to the product of the number of revolutions per minute into the periphery of the crank-pin circle. The conventional measure for the speed of engines, however, is the mean

velocity of the piston, which is equal to the number of revolutions into twice the diameter of the crank-pin circle. The proportion of the mean speed to the maximum speed is therefore as two diameters to the circumference of a circle, or approximately as 7 is to 11; and hence the maximum speed of a piston is equal to its mean speed multiplied by the fraction $\frac{11}{7}$. The ratio of this speed to the allowed maximum speed of the steam in the passages furnishes the ratio of the area of the steam-ports to the area of the piston.

After the area of the ports is thus determined and a suitable length of the ports is selected, their width is found by dividing the area by the length.

It is well to make the length of the ports as large as practicable to procure a small width; for the travel of the valve is directly dependent on this width, and reducing the travel implies a reduction of the amount of work to be expended in overcoming the friction of the valve.

Some attention is to be paid to the dimension of the exhaust port. If it is made wider than necessary the valve must be made longer than otherwise, and a needless increase of pressure and a consequent loss of work from friction will result. If, on the other hand, it is made too narrow, the valve will, at the end of its stroke, cover it more than desirable and choke the exhaust.

In order to find the proper width of the exhaust port we draw the steam-port *S*, (Fig. 11) and over it one leg of the valve with the proper inside and outside lap, first in its neutral and then in

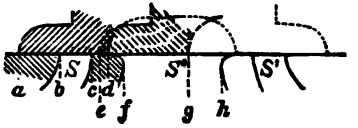

Fig. 11.

its extremest position, removed towards the exhaust port. The distance *ab* is the outside lap, *bc* the width of the port, *cd* the inside lap, and *ae* or *dg* is equal to half the travel of the valve. The width of the bridge, *ef*, should be as narrow as possible, *i. e.* just wide enough to insure a perfect casting and sufficient strength to withstand the steam pressure. But if then the distance *ef* should happen to be less than a quarter inch, the bridge must be made wider accordingly, to prevent the possibility of a leakage of steam from the steam chest to the exhaust passage. If the bridge, on the other hand, is wider than necessary to prevent leakage, the inside corner may be rounded, as indicated in Fig. 11, to facilitate the passage of the exhaust steam. The other edge, *h*, of the exhaust port should have a distance from the inside edge of the valve, *g*, equal to the the width of the steam port *bc*, or not much less in order that the exhaust steam need not contract beyond the area of the steam passage when entering the exhaust port. After the exhaust passage is thus located, the second port can be drawn on the other side and the dimensions of the slide valve can be determined by drawing the other leg of the same over the second port—as shown by dotted lines in Fig. 11.

Velocity of the Valve.

A few words on the velocity of the valve in the different stages of its travel and on the rapidity of the opening and closure of the ports will not be out of place.

The valve moves with increasing velocity from the beginning of its stroke until it reaches its neutral position, where its speed is at the highest rate. Thereafter the speed again diminishes until at the end of its stroke the valve comes, for an instant, to a stop, and then reverses its motion. If we assume a perfectly uniform rotation of the crank-shaft, the velocity of the valve will always be in direct proportion to the leverage under which the eccentric is acting upon the

valve ; and as the selection of a scale for a graphical representation of this velocity is quite optional, we may choose the length of this leverage directly for the representation of this velocity. The maximum velocity will then be expressed by the length, *OI*, (Fig. 3) and this occurs when the eccentric is at right angles, or, in other words, when it passes the line, *OY* When, however, the eccentric is, for instance, at *Γ* the actual leverage is equal to *Oy'*, and the corresponding velocity is expressed by this length. In the foregoing it has been demonstrated that the triangles, *OQk'* and *OΓy'* (Fig. 3) are equal ; and hence *Ok* = *Oy'*. From this it follows that the diagram can also serve to answer the question of the com parative velocity of the valve for any stage of the stroke. All that is needed for this purpose is a line drawn from the point, *Q*, of the diagram, at right angles to the respective crank-line, *OK'*, when the distance of the foot, *k*, of this line, from the centre, *O*, is the equivalent of the velocity of the valve while the crank passes the the line, *OK'*. To find the velocity of the valve at the point of the cutting off, when the crank is at *OE*, (Fig. 12) we draw the line, *Qk* at right angles to *OE*. The distance, *Ok*, then represents the sought velocity,

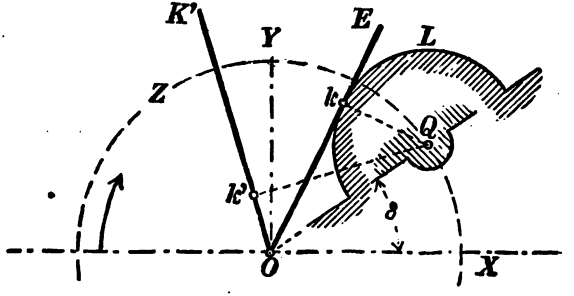

Fig. 12.

and since the point, *k*, is the point of contact of the line, *OE*, and the lap circle, *L*, we may remember, as a rule, that *the distance of this point of contact from the centre, O, always represents the velocity of the valve at the moment of the cutting-off.*

The rapidity of the closure of the port depends not on this velocity alone, but also on the velocitv of the valve from the time it begins until it completes covering the port. Fig. 12 shows that, previous to the cutting off, when the crank is, say, at *K'*, the velocity of the valve, *Ok'* is smaller than *Ok*. This result is consistent with the preceding consideration, according to which the motion of the valve is increasing before reaching the centre of its travel, and as the valve cuts the steam off before that point by virtue of the lap, the velocity is evidently less before the time of the cutting off than it is at that moment. It will also be seen now that the sharpness of the cutting off suffers as the lap is enlarged, and for this, as well as for other reasons, the throw of the valve should be increased as more lap is added.

The length of the steam ports favors the sharpness of the cutting off, and therefore, if like results are desired, the velocity, in other words, the throw, can be lessened in the same ratio as the length is increased. It should, however, at the same time be remembered, that an increase of the length of the ports implies a proportionate decrease of their width, if the same area is to be preserved.

Practical Use of the Diagram.

The diagram explained in the foregoing can be applied in practice to any question relating to the construction of slide valves. The question simply resolves into the problem of drawing the diagram in accordance with the given data, and the sought dimensions or results can be taken from it. Beginners are advised to sketch a diagram of the general form and to mark in it those dimensions that are known or given. The execution of a drawing to scale will then offer no difficulty.

The diagram being drawn, it is easy to proportion the valve gear accordingly. The diameters of the "eccentric path Z" of the diagram represents the travel of the valve, and the throw of the eccentric has to be made to correspond. The proper dimensions of the valve can be found as shown in a previous chapter (width of ports) from the laps given by the diagram. If a valve thus proportioned is properly set, its operation will correspond with the indications of the diagram.

In setting a valve two things are to be taken into consideration: first, the length of the eccentric rod or of the valve stem, and second, the angular position of the eccentric on the crank-shaft. To accomplish the first point it is necessary that either the length of the eccentric rod or else that of the valve stem is made adjustable. It is an object to adjust this length with as few turns of crank shaft as possible, especially on large engines. To this end turn the eccentric until the valve stem is in one its extremest end-positions, say as far *towards* the crank-shaft as it will go. Then adjust the valve according to the greatest port-opening shown in the diagram, or, in other words set it so, that the distance of its outside edge from the outside edge of the respective steam-port (S in **Fig. 1**) is equal to the greatest port-opening of the diagram.

This being done, set the crank on its dead centre with the piston nearest the crank-shaft, and, holding it firmly in this position, advance the eccentric until the proper port (*S* in Fig. 1) is opened by the valve to an extent equal to the lead required, and tighten it on the crank-shaft. If now the crank is turned on its other dead centre, the lead for that side will be found to slightly differ from the lead required, owing to the influence of the angularity of the eccentric rod, which, it will be remembered, we had omitted to consider until now. To correct this difference, the eccentric rod (or the valve rod, as the case may be) must be lengthened by one-half the difference observed; the remaining half of this difference is subsequently removed by changing the position of the eccentric on the crank-shaft, care being taken that the crank remains on the dead centre to which it was adjusted last.

In this way the setting of the valve requires the crank to be set but once to each of its dead centres.

A later chapter will show that sometimes the lead of the fore-stroke is required to differ from that of the return-stroke. The given rule will be equally applicable for setting the valve in such a case.

In the following we shall leave out of consideration the influence of the obliquity of the connecting rod; in other words, we shall consider the connecting rod to be infinitely long. The position of the piston corresponding to any one crank angle may therefore be found by drawing the crank-path to a reduced scale, and projecting the crank pin centre on its horizontal diameter or any other horizontal line. The crank-circle may sometimes be drawn to a scale to make it coincide with the path of the eccentric, as will be shown in some of the following problems.

Problem I. A small steam-engine of 7″ stroke is provided with a valve of $\frac{3}{8}$″ outside lap and $\frac{1}{16}$″ inside lap. The travel of the valve is $1\frac{1}{4}$″ and the lead is adjusted to $\frac{1}{16}$″. What

will be the angular advance, how much expansion and cushioning will take place, when will the exhaust commence, and how much will the outer edge of the valve overrun the inner edge of the port if the width of the port is $\frac{5}{16}''$.

At first we sketch a diagram of the usual form (Fig. 13)

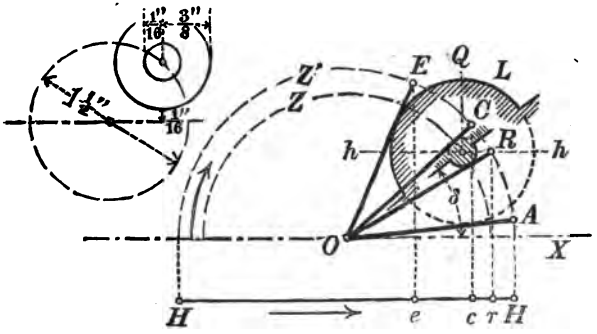

Fig. 13.

and mark in it those dimensions which are known, and in referring to this sketch the course of construction will plainly be seen. The circle *Z*, representing the path of the eccentric, is given by its diameter, equal to the travel of the valve, and can at once be drawn. The lap-circle *L* should be $\frac{1}{16}''$ above the line *OX*, and its radius is known to be $\frac{3}{8}''$, hence its centre *Q* must be $\frac{7}{16}''$ above *OX*. A horizontal line $\frac{7}{16}''$ above *OX* will therefore intersect the circle *Z* in *Q*, and both lap circles may now be drawn. The crank-positions *E*, *C*, *R* and *A*, will respectively indicate the commencement of expansion, compression, release and admission. If we finally draw the crank-circle *Z'* at a reduced scale and project the crank-centres on the line *HH*, which represents the stroke of the piston, we find the cutting-off to take place at *e*, which is $4\frac{1}{4}''$

of the stroke. Cushioning commences at c and the release at r, respectively $\frac{1}{8}''$ and $\frac{1}{2}''$ before the termination of the stroke. The largest port-opening can be directly measured in the diagram and will be found to be $\frac{3}{8}''$ which is $\frac{1}{16}''$ above the width of the port. The angular advance is given by the angle δ. The point of admission is here and in most of the following problems practically at the end of the line *HH*.

Problem II. It is required to find the outside and inside lap of the valve for an engine of 20″ stroke, to cut the steam off at $\frac{3}{4}$ stroke (15″) and to cushion the exhaust steam for 2″. The width of the ports is $\frac{7}{8}''$, the travel of the valve 4″, and no lead is to be given.

The points for cut-off and cushioning being given, the corresponding crank angles can readily be found, and they may be regarded as given, in the sketch Fig. 14. The lead, marked by a dart, should be made $= O$, in other words, the lap circle should rest on the line *OA*. If we draw the crank circle to $\frac{1}{8}$ scale it will coincide with the path of the eccentric.

It may be remarked here that Fig. 14 and most of the following valve diagrams are not made to full size, as they would

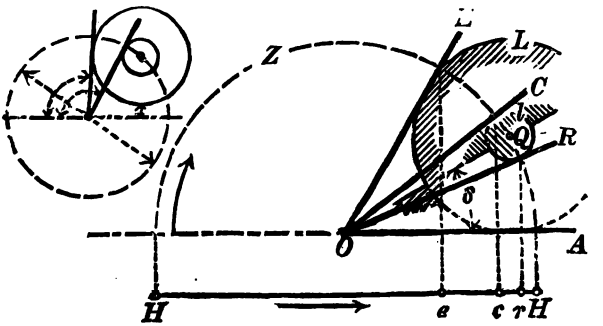

Fig. 14.

take up too much space. In practice, however, the eccentric path should always be drawn to full size to ensure a sufficient degree of accuracy.

The construction will be as follows. Sweep the circle Z with $2''$ radius. Mark on the line HH, representing the stroke of the piston, the points e and c, for cutting off and for cushioning, and obtain, through vertical projection-lines the corresponding crank-angles OE and OC. Both the lines OE and OA must be tangent to the lap circle, the centre of which will therefore be located on the circle Z by a line bisecting the angle EOA. The lap circles L and l can now be swept, from Q, tangent, respectively, to OE and OC, and the crank angle for release, OR, can subsequently be drawn. The radii of the lap circles are the required laps, which are found $1''$ and $\frac{1}{4}''$ respectively. The angular advance is $\delta = QOA$, the release, at r, takes place less then $1''$ before the end of the stroke, and the largest port-opening is $= 1''$, which is $\frac{1}{8}''$ above the width of the port.

Problem III. An engine of $16''$ stroke is required to cut off at $10''$ and have a cushioning of $1\frac{1}{2}''$. The largest port opening is required to amount to $\frac{7}{8}''$ and the lead to $\frac{1}{8}''$.

The crank-angles for cut off and for cushioning, the lead and the greatest port-opening are known, as indicated in the sketch Fig. 15, and the diagram can be drawn as follows:

Sweep the crankpin-circle Z' to reduced scale, and under it the line HH for the stroke of the piston. Mark on it the points e and c for expansion and compression and find the corresponding crank-angles OE and OC. The lap circle must be tangent to OE; it must be $\frac{1}{8}''$ above the line OX and must be $\frac{7}{8}''$ from the centre O. Therefore we draw a horizontal line h $\frac{1}{8}''$ above OX and a circle z, from O, with a radius of $\frac{7}{8}''$. The required lap circle must touch the lines OE, h and the circle z, and its centre can be found by bisecting the angle

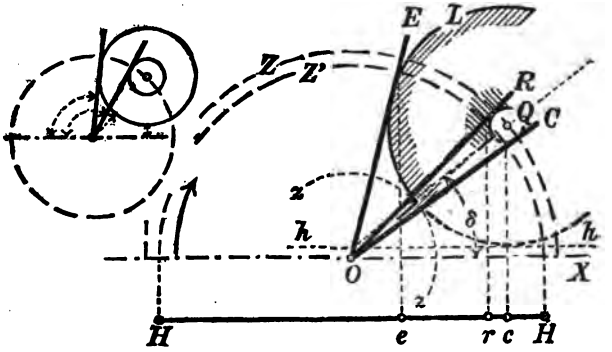

Fig. 15.

formed by OE and h and finding by trial on this line a point Q from which the required lap circle can be drawn. The circle Z swept from O through Q forms the path of the eccentric. A circle from Q tangent to OC gives the inside lap circle, which in this case is found to represent negative lap, since the crank-angle for compression should be tangent to the dotted half of the circle. The necessary dimensions are found to be: travel of the valve $4\frac{1}{4}''$, outside lap $1\frac{1}{4}''$, inside lap, negative, $\frac{1}{8}''$. The release takes place $2\frac{1}{2}''$ before the end of the stroke.

Problem IV. The stroke of an engine is $21''$ the travel of the valve $3\frac{1}{2}''$, the outside lap is $\frac{3}{4}''$ and the inside lap $= 0''$. The width of the ports is $\frac{3}{4}''$ and the lead $\frac{1}{8}''$. By this valve the cutting-off is effected at about $16\frac{1}{2}''$ of the stroke. It is required to change the cut-off to $14''$ of the stroke by changing the throw of the eccentric, leaving the lead angle the same as it was before.

We first draw the diagram for the original travel of $3\frac{1}{2}''$

(Fig. 16) to obtain the lead angle AOX. Next we construct the crank-angle OE for the cutting-off at 14″. Knowing now that the lap circle of the new diagram, the radius of which is

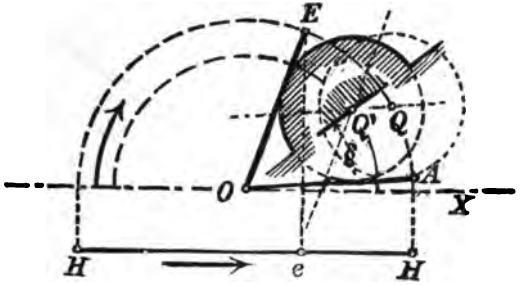

Fig. 16.

to be $\frac{3}{4}$″, must touch both the lines OA and OE, we draw two lines at a distance of $\frac{3}{4}$″ from, and parallel to, these two lines, and the point of their intersection, Q', is the new centre of the lap circle. Its distance from O is the required throw of the eccentric, which is $1\frac{3}{8}$″. The greatest port opening for admission will be $\frac{5}{8}$″ which is $\frac{1}{8}$″ short of the width of the port. The angle of advance, the degree of cushioning and the point of release can subsequently be found from the diagram in the usual way.

The Scope of the Common Slide Valve.

The most primitive form of the slide valve is that without any lap, lead or angular advance, by the operation of which the piston will be exposed to the full steam-pressure on one side, and to exhaust on the other, for the entire duration of the stroke. But the instant the crank passes the centre, the conditions are suddenly reversed and that end of the cylinder which was just being filled with steam is now brought in communication with the exhaust passage. It is evident that all the steam filling the cylinder cannot instantly escape through the opening which at first is very small, and a considerable back-pressure during the first part of the stroke will result. Besides, the expansive power of the steam is not utilized at all, but is prodigally wasted. To obviate these losses lap must be given to the valve and the eccentric must be advanced. But the common slide valve should, under ordinary conditions, never be used to close the ports before half stroke, for a reason that will at once be seen if we examine the operation of a valve with earlier cut-off. Suppose it was required to construct a valve with 4″ travel, to cut-off at $\frac{1}{3}$ of the stroke, with no lead. We draw the circle Z (Fig. 17), determine the crank-angle OE for $\frac{1}{3}$ of the stroke, and bisect the arc EA to obtain the centre Q of the lap-circles. The outside-lap circle can now be drawn at once. Supposing the valve to be without inside lap, the cushioning on the one and the release on the other side of the piston will begin when the crank passes the line OQ. Regarding the line HH' as representing the stroke of the piston we find that compression and release begin at x; the expansion, from e to x, is curtailed by an early exhaust, and the compression from x to H is excessive. If we endeavor to delay the exhaust by giving inside lap, in order to prolong

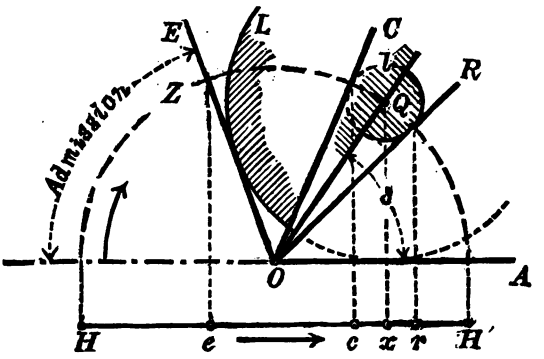

Fig. 17.

the expansive action of the steam, we find, by drawing the lap circle *l* and the crank-angles *OC* and *OR*, that the amount of cushioning will be increased. On the other hand, if we try to diminish the degree of compression it will be necessary to apply negative inside lap, which may again be represented by the circle *l*. The cushioning will then commence at *OR*; but it will be noticed that this measure will hasten the release which then takes place at *OC*. We see from this that with no inside lap, both the release and the cushioning begin much sooner than desirable, and that any change for the sake of delaying the one causes a hastening of the other.

Another very disadvantageous point will be noticed. The greatest port-opening indicated by this diagram is but little more than $\frac{1}{10}$ of the travel of the valve. Hence it is clear that a plain slide valve so proportioned as to cut off at $\frac{1}{3}$ of the stroke produces but a small opening for the admission of the steam.

Irregularities of the Crank Motion.

The relation between the movement of the piston and that of the crank-shaft is in fact somewhat different from that considered in the preceding pages. When the crank is at right angles (see Fig. 18) the piston is not in the middle of its

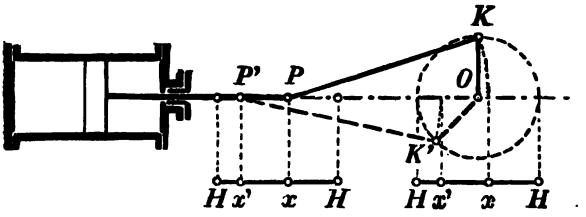

Fig. 18.

stroke but is nearer to the crank-shaft, on account of the obliquity of the connecting-rod. At any other crank-angle the obliquity of the connecting-rod will likewise affect the position of the piston, though to a less extent, the nearer the crank approaches either of its dead centres.

A circular arc swept from the crosshead end of the connecting-rod through the crank-pin centre will intersect the horizontal diameter of the crank circle in a point having the same relation to this diameter as the position of the piston has to its stroke. If it is required to find the true position of the piston for a given crank-angle, instead of projecting the crank-pin vertically, we should project it upon the horizontal diameter by a circular arc; and when the position of the piston is known the corresponding crank-angle can be found in the same way. The radius of this projection-arc should be equal to the length of the connecting-rod, measured on the same

scale to which the crank-pin circle is drawn, and its centre should be located on the line of movement of the crosshead.

These variations from the previously considered ideal movement of the piston will evidently gain in extent as the connecting-rod is shortened, and the latter should, for this as for other reasons, rarely be made shorter than 4½ times the length of the crank. On horizontal engines it is usually equal to 5 times the crank.

It will now be understood that the piston runs ahead of its ideal movement on its "forward stroke" (that is when the connecting-rod is strained on compression,) and stays behind the same during its return stroke. The result will be the admission of more steam on the fore-stroke than on the return-stroke, and hence an unequal development of work at both strokes. This difference will be augmented by the presence of the piston-rod, which diminishes the active surface of one side of the piston, except where an extension of the piston-rod passes through a stuffing box on the back head of the cylinder.

The movement of the valve, in its relation to that of the crank shaft, is subject to similar inequalities, owing to the oscillations of the eccentric-rod, and the valve will be drawn more or less towards the crank-shaft. However, as the eccentric-rod is generally very long in relation to the throw of the eccentric, these variations are small, and besides their effect can easily be neutralized by slightly lengthening the valve stem, which measure will cause the greatest port opening at the fore-stroke to be a trifle less than that of the return-stroke. This correction is achieved by the practice of setting the valve by the lead, or, in other words, by adjusting the valve stem while the eccentric-rod is in an inclined position.

The inequality of the cut-off, occasioned by the obliquity of the connecting-rod, can be rectified by giving the valve a different lap for each of the two ports, but this rectification is

attained at the expense of an equal lead, and is therefore seldom taken advantage of in practice.

An equalization of the compression can be effected by following the same course respecting the inside-lap.

On engines that are provided with valves having equal lap on both sides, a rectification of the point of cut-off can be accomplished by lengthening the valve-stem. This operation is equivalent to an enlargement of the laps relating to a positive movement of the valve and a diminution of those relating to a negative movement; it will therefore destroy the equality of lead, and it will more than rectify release and cushioning. In this case the neutral position of the valve is no more a central one, but is as shown in Fig. 19 where the laps *ab*, and *c'd'* exceed the laps *a'b'* and *cd*, the former pertaining to a positive, the latter to a negative movement.

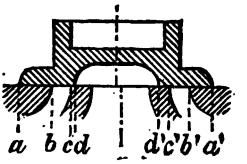

Fig. 19.

Problem V. It is required to find the exact positions of the piston for the cutting off, the release and the cushioning of the engine considered in Problem I., if the connecting-rod is 16″ long.

The diagram can be constructed as shown before, but instead of drawing vertical projection-lines from the crank-pin centres *E*, *C*, and *R*, circular projection-lines are to be used, of a radius equal to 16″, measured, however, on the scale to which the crank-pin circle has been drawn. The resulting points can now easily be transferred upon two

separate lines, for the fore and the return stroke, as shown in Fig. 20.

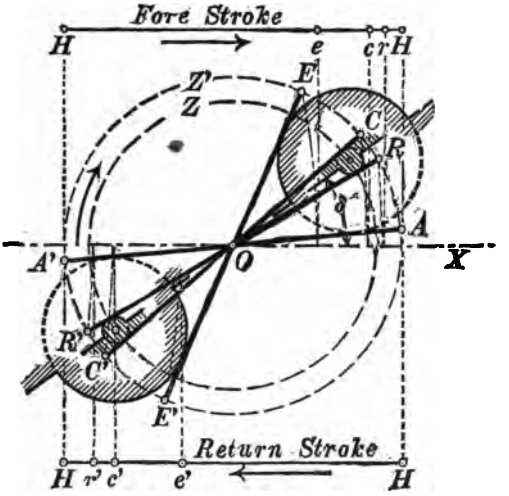

Fig. 20.

A comparison between the fore and the return stroke will be easier if the lower half of the diagram is drawn in combination with the upper half (see Fig. 21). This form of the diagram shows very plainly the amount of the variation, as compared with the results of our first method of projecting by vertical lines.

Problem VI.—The case of Problem II. is to be solved for equalized cut-off and equalized compression, if the connecting rod is 45″ long and the lead for the fore stroke $\frac{1}{32}$″. The solution of this problem involves the drawing of two distinct diagrams, one resembling Fig. 5, the other Fig. 6, both dif-

fering by their lap-circles, since the lap for both ports will differ. Both diagrams can, however, readily be drawn in one figure

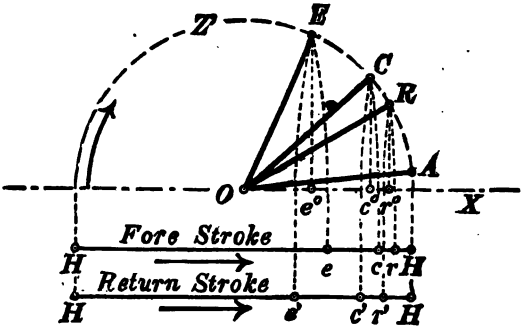

Fig. 21.

We draw in Fig. 22 the crank circle Z, as we did in Fig 14, and mark on the horizontal diameter the points e and e' for the cutting off, and c and c' for the compression. The crank-angles E, E', C, and C' are then found by sweeping the projection arcs of the proper radius, and it will be seen that the corresponding crank-angles for both strokes are no longer diametrically opposite. Since the lead for the fore stroke is given, the diagram for that stroke can be drawn by drawing the horizontal line h, $\frac{1}{32}''$ above OX, and bisecting the angle formed between this line and the line OE, to obtain the point Q on the circle Z, which, it will be remembered, serves in this case for both crank path and eccentric path. By drawing a line from Q through O we find the point Q', which happens to coincide practically with the point C' in this case. Now we can draw the lap-circle L, in full line, from Q to touch OE, and its mate, in dotted line from Q'; also the lap-circle L', from Q' to touch OE', and

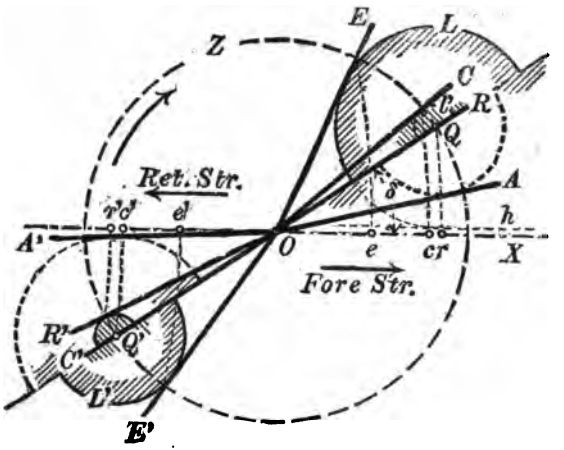

Fig. 22.

its mate from Q. Next we draw, in dotted line (relating to compression, on the exhaust side of the piston), the lap-circle l' from Q, and its mate from Q' in full line. The last lap-circle, from Q' tangent to OC, remains yet to be drawn, but its radius is zero. The diagram now shows that the lead for the return stroke is more than $\frac{3}{8}''$, which considerably exceeds that of the fore stroke, and due attention must be paid to this circumstance in adjusting the valve on the engine. In fact, herein lies the greatest practical difficulty in rectifying the cut-off as shown, since most engineers are accustomed to the practice of setting the valve to equal lead, and this valve, when so adjusted, will distribute the steam to less advantage than if no rectification would have been attempted. In other respects there can be little objection to this measure; for, notwithstanding the considerable lead on one side,

the space which the piston is compelled to travel against the steam after admission is but a very small fraction of the stroke. The crank angles and positions of the piston for the release may be found without difficulty in the diagram.

Those circles swept from Q in full lines represent the laps of that side of the valve that covers the port, *S*, (Fig. 1) while those swept in full lines from *Q'* give the laps for the other side of the valve. The inside lap of the steam passage, *S*, is zero.

Problem VII. The valve considered in problem V is to be rectified for the cut-off by lengthening the valve stem. How much has the valve stem to be lengthened, and what effect will this measure have upon the lead and the exhaust?

The eccentric-path, *Z*, the crank-path, *Z'*, and the point, *Q*, can be copied from Fig. 20. The proposed lengthening of the valve stem will hasten the cutting off at the fore stroke and delay that of the return stroke, and the cutting off will take place at the intermediate piston position, e^0, (Fig. 21) obtained by a vertical projection.* After copying this point from Fig. 21, we can find the exact crank angles, *E* and *E'*, by the projection arcs of the proper radius. The outside lap circles can then be swept, and their distance from the original lap circle equals the length by which the valve stem has to be lengthened, and which is found = about $\frac{1}{16}''$. The lead for the fore stroke will be zero, and that for the return stroke = $\frac{1}{8}''$.

In lengthening the valve stem by $\frac{1}{16}''$ the inside lap for the fore-stroke becomes $\frac{1}{16}'' + \frac{1}{16}'' = \frac{1}{8}''$ and that for the return-stroke = $\frac{1}{16}'' - \frac{1}{16}'' = 0$. We can accordingly finish our diagram and determine the crank angles and piston-positions

*This last assertion is theoretically incorrect, for the point of cut-off, after the proper rectification will not precisely coincide with the point, e^0, but the difference is practically inappreciable.

for cushioning and release, and a comparison of the resulting points c, c', r and r' with the corresponding points of Fig. 20 will show that the measure resulted in an over-correction of release and cushioning.

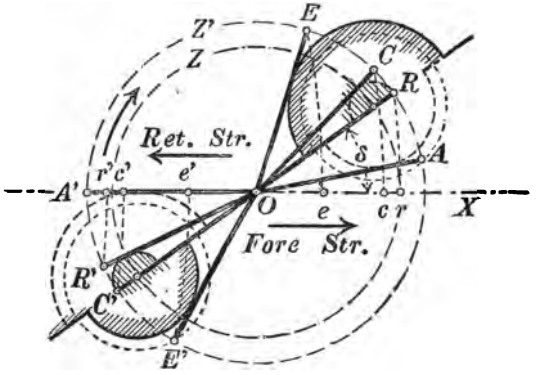

Fig. 23.

PART II.

LINK MOTIONS.

Link Motions.

It is often required that steam engines should be constructed to run either forward or backward, as is the case with locomotives, marine engines, hoisting engines and others. Valve gears, by which this end can be accomplished, are called "reversing gears," and most of those now in practical use admit of a variation of the degree of expansion. The almost exclusive use of "Link Motions" for this purpose makes it of the utmost importance that steam-engineers should be perfectly familiar with the peculiarities of these most ingenious devices, especially since they can be proportioned to correct the difference between fore and return stroke occasioned by the obliquity of the connecting-rod.

The number of different link motions and other reversing gears admitting variable expansion is not small, but it seems that the two first inventions (the Stephenson, and the Gooch link motion) have stood the test for fitness and durability better than any of their followers. We shall therefore consider the theory of these two gears only, but it may be remarked that the operation of nearly every other reversing-gear can, in some measure, be compared with that of either of these two link motions.

The Stephenson Link Motion.

This link-motion, the general arrangement of which is shown in Fig 24, is the one most generally used, especially on locomotives.

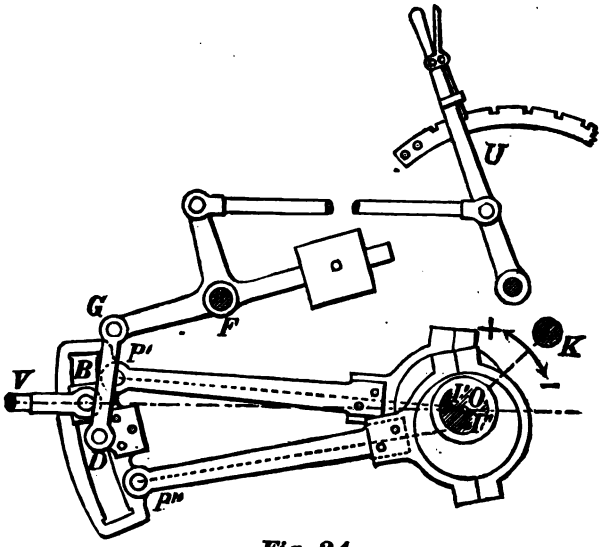

Fig. 24.

Two eccentrics P and P'' are secured to the crank-shaft, and the ends P' and P'' of the rods of these eccentrics are jointed to the "expansion link" or commonly called, briefly, "link," which is a curved frame of wrought iron or steel, containing a sliding block B that is fitted into the slot of the link and is attached, by a joint, either directly to the valve-stem V or else to a rocker by which the slide-valve is moved.

Two of the more usual forms of the link are shown in Figs. 25 and 26. The slide-valve is a single one of substantially the same shape as is used in stationary engines. The link is suspended at the stud *D* (Fig. 24) by the suspension-rod, or "hanger," *GD*, and can be lowered or raised by means of the "suspension lever" *FG*, which is connected, as the drawing shows, to the reversing-lever *U*, the latter moving on a notched arc on which it can be fixed at any desired position. The hanger *GD* may be attached to the lower end of the link (see *D* in Fig. 25 or *P''* in Fig. 26) or, to a stud secured to the centre of the link by a bridge or bracket (Fig. 24.) Sometimes the suspension lever *FG* is located below the link, placing the hanger in an inverted position.

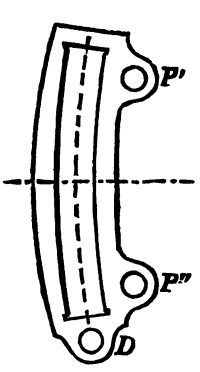

Fig. 25.

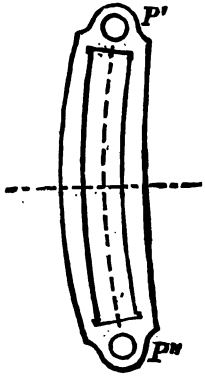

Fig. 26.

The eccentric *I'* is called the forward eccentric, and is fastened to the crank-shaft in such a position that its operation upon the link-block *B* would produce a forward-rotation of the engine. The other eccentric, *I''*, is the backward or backing eccentric, and would cause an opposite rotation of the engine if conected with the link-block.

When the crank-shaft rotates, these two eccentrics will transmit to the link a movement of a very peculiar character, being both oscillating and reciprocating at the same time. By lowering or raising the link either of the two eccentrics can be brought in action, and thus the engine may be run forward or backward at pleasure. The link can, however, also be used in intermediate positions or "grades" when both eccentrics more or less influence the movements of the valve, and it will be the chief object of our future investigation to find the law of the movement of the valve for any grade of the gear.

The eccentric-rods may be used either "open" or, "crossed." They are called "open" when they are connected with the link as represented in Fig. 27, and "crossed" when they are arranged as Fig. 28 shows. To make this distinction both eccentrics must be inclined *towards the link*.

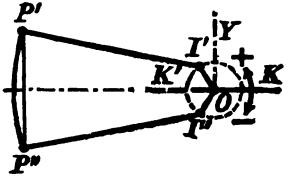

Fig. 27.

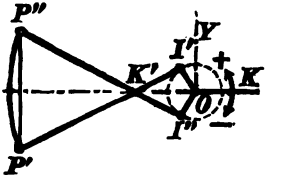

Fig. 28.

for when the crank-shaft is turned half a revolution, to bring the eccentrics on the other side, the link-motion with open rods will appear crossed, and *vice versa*, and care must be taken, therefore, not to confound both types.

When the link-block is not attached directly to the valve-stem, but to a rocker, by which the direction of motion is reversed and so transmitted to the valve, the position of the crank in relation to the eccentrics is indicated by the lines

OK' (Fig. 27 and 28) as will be understood by consulting the rules given in the chapter on "the angle of advance."

A right line drawn from the centre O of the crank-shaft through the block B shall be termed the "centre-line" of the link motion. When the crank is on one of its centres, the centre lines of the eccentrics I' and I'' may form equal or unequal angles with the line OY, which is drawn perpendicularly to OB. The reason why these angles are sometimes made unequal will be given later.

The Theory.—The described link motion consists of a number of oscillating and reciprocating parts, and the exact law of the movement which the link in its several grades of gearing transmits to the slide valve is extraordinarily complicated. Our theoretical investigation will therefore be based upon a series of assumptions similar to those we made when considering the common slide valve. We shall thus disregard the effect of the oscillation or obliquity of the connecting rod and of the eccentric rods, and, besides, assume that every point of the link were moving in a right horizontal line, or, in other words, we shall overlook the vertical vibration of any part of the link motion. A subsequent chapter will then treat of the modification that our theoretical result suffers from these neglected influences, and show a method for finding the means to make these influences compensate each other.

The mentioned assumptions would imply a distortion of the link during its movement, and to keep these distortions within as small a margin as possible, we shall study the movement of the link while in its mid-gear, when the link block will be operated by the centre of the link. It can then be demonstrated that the movement of any point of the link can be considered as though it were produced by a separate eccentric, and it remains now for us to find the position of this imaginary eccentric.

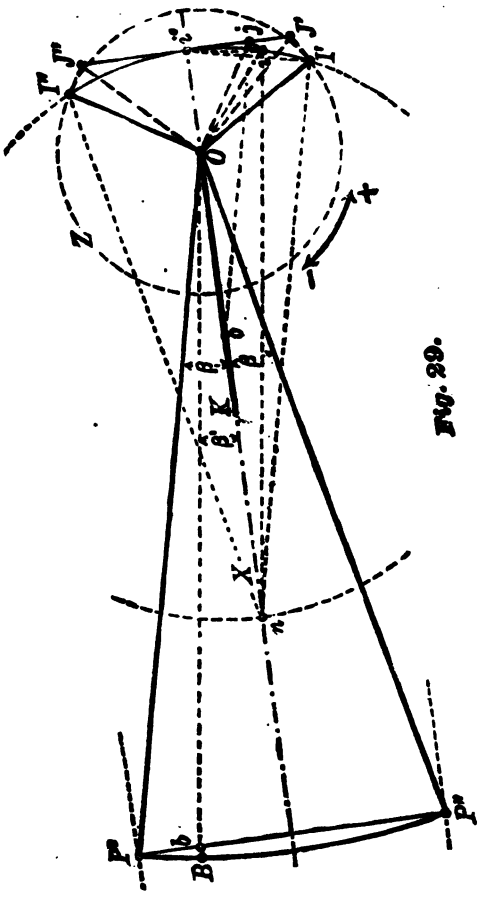

FIG. 29.

Suppose I' and I'' (Fig. 29) to represent the two eccentrics of a link motion, the link $P'\,P''$ of which is shown in its neutral position, obtained by transferring the ends of the eccentric rods to the centre O of the crank shaft. Since we assume that all points of the link, and hence also the pins P' and P'', were moving in horizontal lines, we should first understand the movement of the latter. Let us consider for this purpose the forward eccentric separately, in Fig. 30.

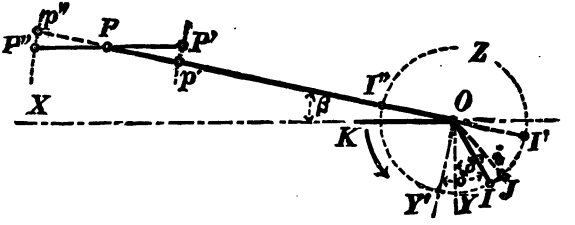

Fig. 30.

If the end P of the eccentric rod in this case were moving in the line $p'\,p''$, the angular advance of the eccentric I for this movement would be the angle $Y'OI$, OY' being at right angles to OP. If it would be desired to produce the same movement on the horizontal centre line (by a horizontal eccentric rod), the proper position of the eccentric would then be found at i by making its angle of advance $YOi = Y'OI$. However, the movement of the pin P is actually in the line $P'\,P''$, and, therefore, when the eccentric I is at I' or I'' the end P will not be at p' or p'', but at P' or P'', respectively. The arcs $p'\,P'$ and $p''\,P''$ are very short in proportion to their radius, and may therefore be considered, approximately, to be right lines, forming right angles with $O\,P$. Now it will be observed that the travel $P'\,P''$ slightly exceeds $p'\,p''$, and hence also the diameter of the eccentric path Z, and to

produce a movement, on the horizontal line $O\,X$, equivalent to that of the pin P on the line $P'\,P''$, we would have to increase the throw of the eccentric from i to J, by making $O\,J = P\,P'$. The point J can therefore be regarded the "ideal eccentric" for the movement of P.

A glance will now show that the angle $I\,O\,J$ is equal to the angle $P\,O\,X - \beta$, in other words, to obtain the ideal eccentric J, *the eccentric I would have to be moved on the crankshaft, through the same angle and in the same direction as the eccencentric rod P would have to be moved to bring it in a horizontal position.* Besides, the triangle $O\,I\,J$ is equal to the triangle $P\,p'\,P'$, whence it appears that *the line J I is at right angles with O I.*

Applying these results to Fig. 29, we find the points J' and J'' representing the ideal eccentrics for the movements of the extreme points P' and P'' of the link, by making $I'\,O\,J' = I''\,O\,J'' = \beta$, and besides $I'\,J'$ and $I''\,J''$ at right angles to $O\,I'$ and $O\,I''$, respectively.

We are now prepared to study the movement of any point of the chord $P'\,P''$ of the link, say of b. It can be asserted that, by dividing the line $J'\,J''$ in the same proportion in which the point b divides the chord $P'\,P''$, a point j is obtained that can be considered the ideal eccentric of the movement of the point b. (The point b being on the forward end, P', of the link, the point j is on the side of the forward eccentric J' of the line $J'\,J''$.) To prove this proposition we show, in Fig. 31, the crank $O\,K$ rotated through the angle a, and also the several real and ideal eccentrics, in their corresponding position. The link will then assume the position $P'\,P''$, and $p'\,p''$ shows its neutral position, intersecting the line $O\,X''$ in the point m. We then project, upon the horizontal centre line, the points P', P'' and b and the ideal eccentrics $J'\,J''$ and j, obtaining the points X', X'', X, x', x'' and x.

According to the foregoing demonstrations, the position

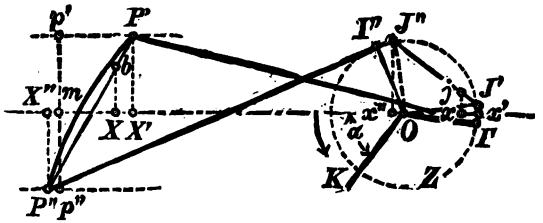

Fig. 31.

of the points x' and x'' in relation to O will correspond with that of the points X' and X'' in relation to m, and since the position of x in relation to x' and x'' will correspond with the position of X in relation to X' and X'', on account of the proportional division of the lines $J'J''$ and $P'P''$, it is evident that the distances Ox and mX are equal; or the horizontal movement of the point j equals the horizontal movement of the point b, and our assertion is demonstrated.

It must, however, be remembered that an eccentric at j can produce a movement as said only by means of a horizontal eccentric rod, and if it was desired to produce the movement of the point b by a direct connection, it would be necessary that the eccentric rod should have an inclination $-\beta'$ (see Fig. 29). The case would then come under the same head as that considered before (see Fig. 30), and knowing the ideal eccentric j (Fig. 29) we can find the desired eccentric z by making the angle $jOz = \beta'$ and jz at right angles to Oz. Thus it is demonstrated that if an eccentric, at z, were added to the two eccentrics I' and I'', and were connected by an eccentric rod with the point b of the link, these three eccentrics would operate in perfect harmony; but it should never be lost sight of, that this entire demonstration is based on our initial assumptions and is therefore only approximately correct.

A lowering of the link that brings the point b in the centre line $O X$ will also bring the imaginary eccentric i in a direct line of action, and the link-block will move as if it was moved by this eccentric; the point i is therefore the ideal eccentric of the movement of the valve for this grade of the link.

The movement of other points of the link can be examined in the same way, and the result will be a series of "ideal eccentrics," like i, forming a continuous curve $I' i I''$ (Fig. 29) which may be termed "*locus* of ideal eccentrics," or briefly *locus*, and it is demonstrable that this curve is a circular arc, tangent to the line $J' J''$.

Directing our attention to the quadrangle $O i^{\circ} j i$, we will notice that the angles at i° and at i are right angles, and therefore, if we draw the line $i i^{\circ}$, the angles $i i^{\circ} j$ and $i O j$ will be equal, according to a well known geometrical proposition. This shows that the line $i i^{\circ}$ forms right angles with the line $O b$, since we made the angle $i O j = \beta'$, which is the angle through which the link and the eccentric rods have to be lowered (starting from the mid-gear) to bring the point b, and hence also the ideal eccentric i, in direct action. The same thing is true for every other ideal eccentric, and also for the real eccentrics I' and I''.

Referring to this fact, we can find the centre of the "*locus*" by drawing a line through I', perpendicularly to $O P'$, that intersects the centre line $O X$ in i°, and by subsequently bisecting the line $I' i^{\circ}$ by a line parallel to $O P'$, that intersects the line $O X$ in the required centre o of the arc $I' i^{\circ} I''$.

If now the circle of the *locus* is extended to its second intersection at n, with the line $O X$, and we draw the lines $n I'$ and $n i$, it can easily be proven that the horizontal inclination of these lines will equal the angles β and β' respectively; hence $n I'$ is parallel to $P' O$ and $n i$ parallel to $b O$. This fact can very conveniently be used for locating on the *locus* the ideal eccentric for any point of the link.

In the foregoing we investigated the movement of a link with open rods. If, however, the rods are crossed, the inclination of the eccentric rods will have the effect of diminishing, not increasing, the angular advance of the eccentrics. The ideal eccentrics for the movement of the ends P' and P'' of the eccentric rods (Fig. 32) will be found to be located at J' and J'', in accordance with the rule found in relation to Fig. 30. The *locus* of ideal eccentrics can now be determined as before, and will again be found to be a circular arc, but its convex side will be turned towards the centre O of the crankshaft. The rules for finding the centre of this arc and for locating the ideal eccentric for any point of the link, are the same for crossed as for open rods.

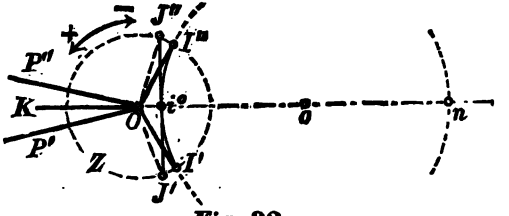

Fig. 32.

If the link has the form shown in Fig. 26 it cannot be lowered or raised far enough to bring the pins P' and P'' in the centre line, and the extreme positions to which the link can be lowered or raised were heretofore regarded intermediate grades. Only a portion of the link is available, and we can therefore use only a portion of the *locus* of eccentrics, as shown in Fig. 33.

For convenience, the rule for constructing the *locus* of ideal eccentrics shall here be recapitulated in a condensed form.

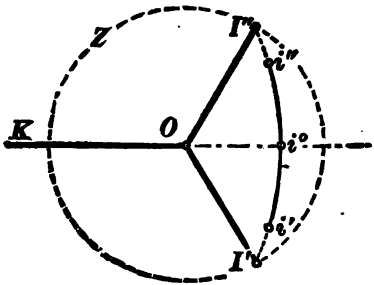

Fig. 33.

Make a skeleton drawing (Fig 29) showing the position of the eccentrics I' and I'', relatively to the centre line of the link motion $O\,X$, while the crank is on one of its dead centres. Then draw the link $P'\,P''$ in mid-gear, in its neutral position (which may be done to reduced scale) and a line through the forward eccentric I', at right angles with the forward eccentric rod $O\,P'$, meeting the line $O\,X$ in i°. Bisect the line $I'\,i^{\circ}$ by a line parallel to $O\,P'$ and obtain the centre o of the *locus*.* Sweep the *locus* $I'\,i^{\circ}\,I''$ and mark at the same time the point n of the same circle. The ideal eccentric of any point of the link, say of b, is found by drawing through n a line parallel to the line $b\,O$, which intersects the *locus* in the sought point i.

The Valve Diagram.

According to the foregoing, the movement of the valve of the Stephenson link motion can be supposed to be derived from a series of eccentrics, and hence the distribution of the steam can be studied by a series of valve diagrams. To find them we perform the double operation of rotating the diagram of ideal

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* If the eccentric rods are crossed, the points P' and P'' will exchange their places, and the centre o of the *locus* will accordingly be found on the other side of the crank-shaft.

eccentrics ($I'i°I''$) through 90° and inverting it. This transfer or conversion can, in fact, most readily be accomplished by means of a piece of tracing paper, in the manner described on page 11. Thus we obtain the curve $Q'\,q°\,Q''$ (Fig. 34) on which the centres of the lap circles of the required valve diagrams will be located.

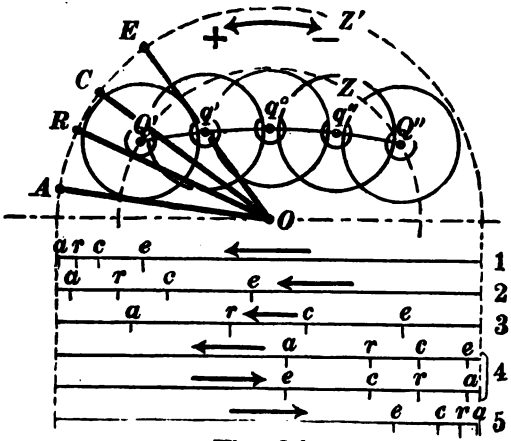

Fig. 34.

The investigation of five different grades will fully suffice for a clear understanding of the operation of the link, and the study of more intermediate grades will offer no difficulty. We shall choose for this purpose the lowest position of the link and every following grade produced by successively raising the link one-quarter of its length, ending with the link in its highest position. The corresponding centres of the valve diagram (Fig. 34) are Q', q', $q°$, q'' and Q''. Both lap circles can now be swept from each of these points, and we obtain for the first grade, forward, the crank angles OE, OC, OR and OA for expansion,

cushioning, release and admission of steam. After drawing the crank circle Z', to reduced scale, we can find the corresponding positions of the piston, e, c, r and a on the line 1, by projecting the points E, C, R and A. The positions of the piston for the corresponding stages of the steam distribution of the other grades can be located in the same manner, and are marked on the lines 2, 3, etc. The crank-lines necessary for their construction are, however, omitted in Fig. 34, to avoid confusion. At all these grades the steam is admitted, on the steam side, until the piston arrives at e, it expands between e and r, and is exhausted for the rest of the stroke. The exhaust side of the cylinder is in communication with the exhaust passage before arriving at c, where cushioning begins, and at a the port is opened again and steam is admitted for the following stroke.

Now, it will be seen that at grade 2, expansion, compression, release and admission take place earlier than at the first grade, showing that the link furnishes means for varying the degree of the cut-off. The action of the valve at the mid-gear, or grade 3, is remarkably symmetrical as regards both ends of the cylinder. Expansion, compression and release begin at very early stages of the stroke, and the admission of steam for the following stroke, at a, takes place at such an early time that the piston has to force the admitted steam back into the boiler to complete its stroke. This action will last as long as the admission of steam at the beginning of the stroke, and the work then developed will be entirely consumed. Steam is exhausted on the exhaust side of the cylinder from the beginning of the stroke up to c, where cushioning begins, while on the other side steam is exhausted after the piston passes the point r. The steam expands while the piston moves from e to r, and the exhaust steam is compressed between c and a, which are again at equal distances, but as the mean pressure of the expanding steam generally exceeds the mean pressure at compression, there is a slight gain of work, which, however, is rarely sufficient to run

the engine when without any load. This singular process is the same whether the crank-shaft rotates in the one or in the other direction. By reason of these facts the centre of the link is called, very properly, the "dead point of the link."

Passing over to grade 4 we find the action of the valve to be hastened still more, (see upper line.) The steam is now admitted on the exhaust side (after the piston passes the point *a*) for a considerably longer period than on the steam side, (before the piston passes the point *e*); hence the motion of the crankshaft will be retarded—the admitted steam tends to rotate the shaft *backward*. At this grade the link is indeed raised so far that the backward eccentric I'' exerts a predominating influence upon the valve, and we might have arrived at the above conclusion without the diagram. The distribution of steam will, therefore, take place as shown by the lower line. However, the upper line, (grade 4) shows the process when a running engine is suddenly reversed. The engine will act like a pump, forcing air from the exhaust passage into the boiler until a stoppage and reversal of the rotation is effected. But as most locomotives exhaust into the smoke-stack, the air which is thus pumped into the boiler contains ashes in suspension which are apt to precipitate on the walls of the cylinder. It is, therefore, very injurious for the engine to reverse in this way, unless provision is made to cut off the communication of the exhaust-passage with the smoke-stack.

The line 5 (Fig. 34) shows the action of the valve when the link is fully reversed.

As regards the lead, the behavior of the Stephenson link-motion is very peculiar. The lead being represented by the distance of the lap circle from the horizontal centre-line, the diagram (Fig. 34) indicates that the lead is variable at the different grades. It is greatest at the intermediate grades and least at the full-stroke grades. It will further be seen that the lead at the mid-gear is equal to the greatest port opening of the

same grade, the valve being at the end of its travel when the crank is on its centre. The greatest port opening is found to be greatest at the full-stroke grades and least at mid-gear.

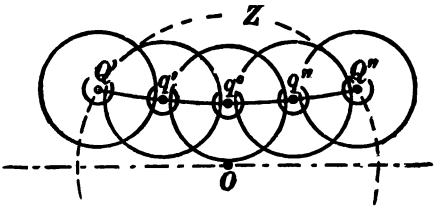

Fig. 35.

In Fig. 35 is shown the valve diagram of a link-motion with crossed rods. We find in this case the behavior, relating to the lead, reversed, as compared with Fig. 34, and besides the greatest port openings for intermediate grades decidedly smaller. This latter is the principal reason why crossed rods are seldom in use.

The Link and Its Suspension.

The slot of the link should be curved in such a way that the "centre of movement" or "neutral position" of the valve should be the same for every grade of the link. Hence, if we transfer the ends of the eccentric rods to the centre *O* (Fig. 36) of the crank-shaft a lowering or raising of the link should not affect the valve. This condition is fulfilled when the curvature of the link (in the position shown in Fig. 36) is a circular arc swept from the centre *O* of the crank-shaft. Both eccentric rods should, therefore, be of equal length, and the radius of the centre line of the slot

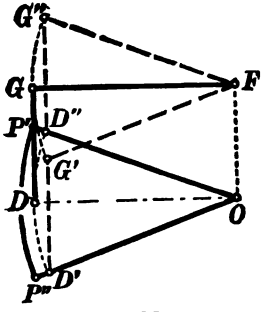

Fig. 36.

Curve of Link

equal to the length of the eccentric rod plus the distance of the pin-hole P' (Fig. 25) from the centre line of the slot. If the link is as shown in Fig. 26, the latter distance is = zero.

The link should be made neither too long nor too short. A definite rule cannot be given to comply with the different conditions to which link motions may have to be adapted. Good results may be obtained, in most cases, by making the available length of the slot of the link about six times the throw (OI') of the eccentrics. The actual length, of course, must be longer by something more than the height of the link-block.

The mode of suspension of the link has a decided influence upon the regularity of the action of the link-motion. In our investigation we assumed that every point of the link were moving in a horizontal line. To approach this supposition as near as possible, the suspension rod GD (Fig. 36) should be made as long as space and convenience will permit, in order to flatten the arc in which the point D moves. Besides, to keep this arc in a horizontal position, the neutral position of the rod GD should invariably be vertical, whether the link be raised or lowered, and this will be the case if the upper point G of that rod is guided in an arc $G'\ G''$, which is equal to the arc $D'\ D''$, in which the pin D of the link is moving as the link, in its neutral position, is lowered and raised. The centre F of the suspension lever should, therefore, be vertically above O, at a distance equal to the length of the hanger GD, while its length FG, should be equal to the distance OD in Fig. 36.

If the link is suspended by the lower pin P'', Fig. 37 will have to be substituted for Fig. 36, in the above demonstration. In practice, the length of the suspension lever, FG, is seldom made of the length required by this rule, sometimes for want of space, at other times on account of the inconvenience of such an extreme length. Besides, it is a general practice to purposely introduce certain irregularities by the mode of suspending the link, to compensate for the unequality between

fore and return stroke, caused by the angularity of the connecting rod. This matter will be discussed later.

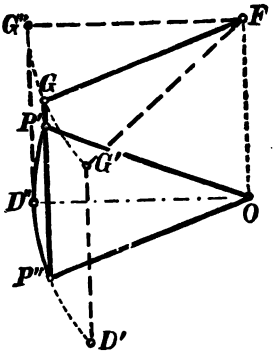

Fig. 37.

Equalization of the Lead.

We have seen that the lead of the Stephenson link motion is variable with the different grades, and this is regarded by some engineers as a disadvantage of this valve gear. It is, however, questionable if this objection is really worth much attention. We shall, therefore, but briefly discuss the method that is sometimes employed to equalize the lead.

The said variation of the lead is owing to the change of the inclination of the eccentric rods in changing the grade, and will, therefore, be reduced by the use of long eccentric rods and a comparatively short link. It can be traced back to the variation of the locus $I'i^{\circ}I''$ (Fig. 29) from a vertical line.

The method alluded to consists in giving to the forward eccentric I' a greater angle of advance than to the backward eccentric I'', (Fig. 38), when the locus of ideal eccentrics can

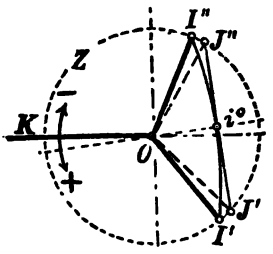

Fig. 38.

be found as before, the result being the arc $I'i°i''$. The lower half of this curve now approaches more nearly a vertical line, (compare Fig. 38 with Fig. 29), showing that the lead will be less variable at the different forward grades, and this variation is least if the point $i°$ is vertically above I'. The upper half of the locus in Fig. 38, however. deviates considerably more from a vertical line than before, hence the equalization of the lead for the forward grades is accomplished at the expense of the lead of the backward grades.

A link-motion with equal angles of advance can be changed to have equalized lead at the forward grades by advancing both eccentrics through a certain angle, in the direction of the forward rotation of the shaft, or else by moving the crank in the opposite direction through the same angle. In this latter case, nothing in relation to the link-motion is disturbed, and the original diagram of ideal eccentrics (Fig. 39) will serve, the only difference being a change of the crank from OK to OK', and the latter line is the base line for the new diagram, as regards the angles of advance, while OK remains the centre line of the locus. This result is identical with that obtained before in Fig. 38, but by this way of reasoning we can easily ascertain through what angle the crank should be shifted in order to get the line $I'i°$ at right angles with the new base line

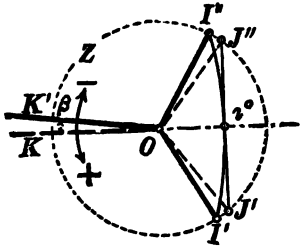

Fig. 39.

OK'. This angle is evidently $= I'i^{o}J''=\beta$. (See Fig. 29.) If the link is as shown in Fig. 40, the crank need only be shifted so that the points i' and i^{o} (Fig. 33) are in a vertical line, and hence the angle β' (Fig. 40) should be used for the correction.

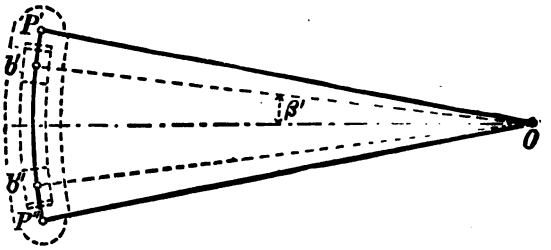

Fig. 40.

The corresponding valve diagram is shown in Fig. 41, where it will be seen that the lead remains nearly the same for the grades Q' q' and q^{o}, but it diminishes rapidly, as the link is raised farther and adjusted to the grades q'' and Q''. At the latter grade the lead may even become negative, in other words, the opening of the steam-port takes place after the crank passed its centre.

This method can evidently be applied with advantage only on engines which are principally used running forward, as is the case with marine engines and locomotives.

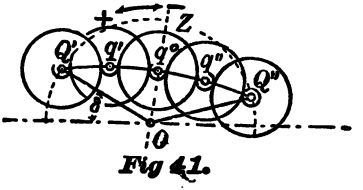

Fig 41.

PROBLEMS.

Problem VIII.—A Stephenson link-motion, with open rods, is to be constructed, to cut off at $\frac{4}{5}$ of the stroke when at full grades, with a lead of $\frac{1}{8}''$. The largest port opening for the full grades is required to amount to $1\frac{3}{8}''$ and the inside lap to be $\frac{1}{8}''$. The link is of the form shown in Fig. 25, its length $P'P''$ is $12''$, the distance of the pins P' and P'' from the centre-line of the slot is $3''$ and the length of the eccentric rods is $3'\ 6''$. The link-block is directly geared to the valve-stem, and at full grades the eccentric rods are to be in line with the link-block. The proper dimensions and positions of the valve and eccentrics are required, and besides the elevation of the link when required to cut off at $\frac{3}{5}$ and $\frac{2}{5}$ of the stroke.

Since we know the greatest port opening, the lead, the point of cut off and the inside lap, for both full grades, we can find the points Q' and Q'' of the valve diagram (Fig. 42). Next we

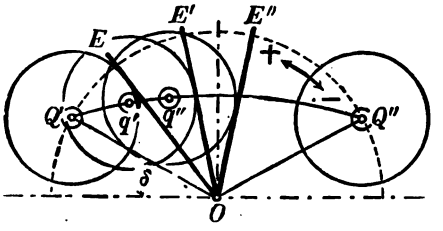

Fig. 42.

locate the link $P'P''$ (Fig. 43) in its neutral position in mid gear at reduced scale. Mark the eccentrics I' and I'' to correspond with Q' and Q'' of the valve diagram, draw the line $I'i^o$ at right angles to $P'O$ and bisect $I'i^o$ parallel to $P'O$ to get the centre o of the locus. The locus can now be drawn in Fig. 43 and transferred to Fig. 42. In Fig. 43 the point n can also be marked, either by extending the circle of the locus, or by drawing $I'n$

parallel to OP'. To find the positions of the link for the required intermediate grades we draw the crank angles OE' and OE'' (Fig. 42) to correspond with $\frac{3}{5}$ and $\frac{2}{5}$ of the stroke of the piston, and find on the curve $Q'Q''$ the centres q' and q'' of the lap circles that are tangential to these crank angles. Finally we transfer the points q' and q'' to Fig. 43 (i' and i'') and draw

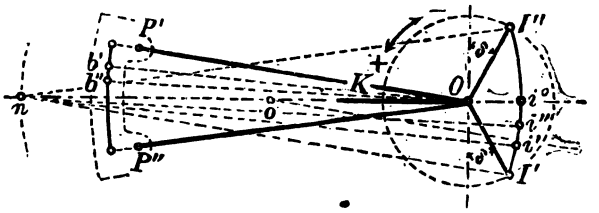

Fig. 43.

the lines Ob' and Ob'' parallel to $i'n$ and $i''n$, and b' and b'' are the points of the link for the required intermediate forward grades; those for the corresponding backward grades are in a symmetrical position. We find, besides the angle of advance $=\delta$, the throw of the eccentrics $OI' = OI'' = 2\frac{3}{8}''$ the outside lap $= 1''$ The radius for the centre-line of the slot of the link should be $= 3'6'' + 3'' = 3'9''$.

Problem IX. A Stephenson link-motion with open rods is required with a link of the form shown in Fig. 26. The cutting off for the first forward grade is to take place at $\frac{4}{5}$ of the stroke, with a lead of $\frac{1}{4}''$. The largest port opening is to be $= 1\frac{3}{8}''$ and the inside lap $\frac{1}{8}''$. It is desired that the lead should be equalized for the forward grades. The length of the eccentric rod is $3'6''$, the link-block is attached, at a leverage of $9''$, to a rocker, from which the motion is transmitted to the valve by an arm of $12''$ length. The centre line of the engine is inclined to the centre-line of the link-motion, by an angle of $15°$,

as shown in Fig. 44. The length of that portion of the link ($b'b''$ Fig. 40) that is actually used is to be equal to three times the travel of the link-block when in full gear. It is required to construct the valve diagram, to find the throw and the position of

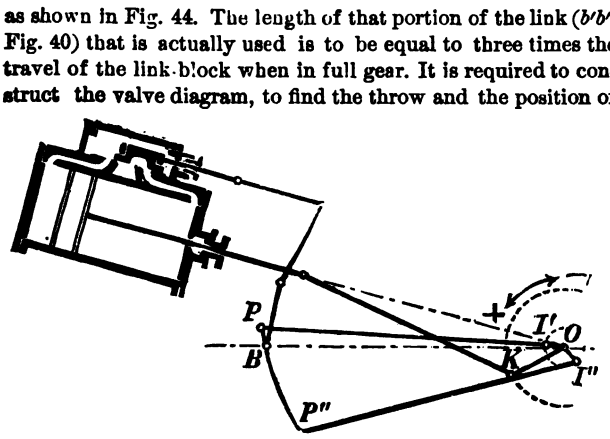

Fig. 44.

the eccentrics, and the length of the link shall be determined, if the distance $I'b'$ (Fig. 40) is $=3''$. Any question relating to intermediate grades can, of course, be solved as in the preceeding problem.

The required valve diagram is shown in Fig. 41. At first we can construct the centre Q' for the first forward grade, showing the travel of the valve at that grade to be $4\frac{1}{2}''$ with an outside lap of $\frac{1}{8}''$. Owing to the unequal leverage of the rocker, the corresponding travel of the link block is $3\frac{3}{8}''$ and the length $b'b''$ of the link is accordingly $10\frac{1}{8}''$. The length of the eccentric rods being given, we can now locate these two points of the link, at reduced scale, (Fig. 45) and find the angle β', which is the angle of equalization of the lead. We can thus consider the case in the way shown in Fig. 39, *i.e.*—we consider the line Ob' (Fig. 45) as the base line of the diagram of eccentrics, and can accordingly transfer the point Q' (Fig. 41) of the valve

diagram to its proper position i' in the diagram of eccentrics Fig. 45, by the angle δ. The locus can next be drawn, as usually, and it should be extended to both sides, in this case. The drawing of the link can then be completed by making $b'P' = b''P'' = 3''$, and the points I' and I'', corresponding to the points

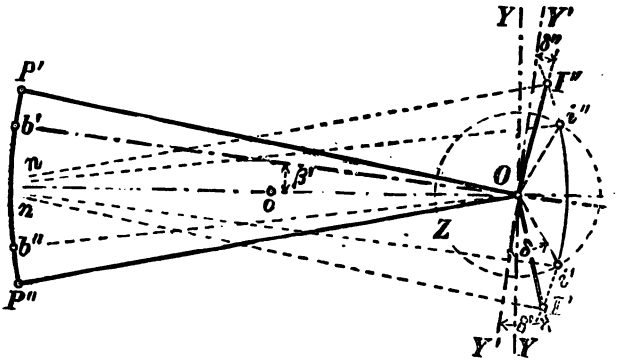

Fig. 45.

P' and P'' are found by making nl' and nl'' parallel to $P'O$ and $P''O$, respectively. But for the unequal leverage of the rocker the points I' and I'' would represent the eccentrics by which the points P' and P'' of the link should be worked, and their throw would be $3\frac{1}{4}''$. But as the case is, this figure is subject to a reduction and the actual throw of the eccentrics must be made $2\frac{7}{16}''$.

When we complete the valve diagram, Fig. 41, we find the lead for the first backward grade to be negative, about $\frac{1}{4}''$, which is certainly not a very desirable feature. The variation of the lead in the present link-motion assumes such proportions, on account of the rather unusual shortness of the eccentric rods.

The slot of the link should be curved by a radius of 3'6", equal to the length of the eccentric rods.

To find the proper position of the eccentrics on the crank shaft we determine the angles δ' and δ'', the line OY' forming right angles with the base line $\delta' O$ (Fig. 45). Then we draw the crank OK (Fig. 46) on its centre, which is at an inclination of 15°, and locate the eccentrics I' and I'' by their angles of advance, δ' and δ''.

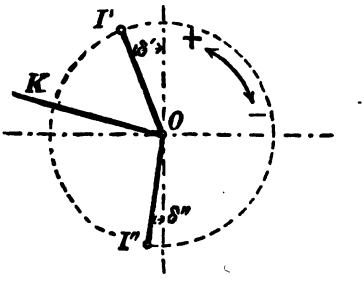

Fig. 46.

IRREGULARITIES AND THEIR RECTIFICATION.

The results of the foregoing considerations will be somewhat modified when we take into account the complicating elements of the link and crank motion.

The effect of the angularity of the connecting rod has been considered before.

The irregularties arising from the oscillations of the eccentric rods can be corrected by adjusting the valve to have equal lead at the fore and return stroke.

The oscillation of the link has the effect of drawing the ends of of the eccentric rods nearer the centre line, and since this (by reason of the inclined position of the eccentric rods) is accompanied by a horizontal movement from the crank-shaft, this influence is equivalent to a temporary lengthening of the eccentric rods while the link is inclined.

Another source of variation from our theoretical results is the "slip." Owing to both the mode of suspension and the oscillation of the link, the latter is caused to slip up and down on the block, and the valve is not operated from a fixed point of the

link. The slip depends mainly on the mode of suspension and, as a general rule, has likewise an effect equivalent to a temporary lengthening of the eccentric-rods as the link assumes an inclined position. The slip can be reduced by lengthening the link and by suspending it as shown in Fig. 36 or 37.

There are still other influences of a similar character, but being of minor importance their special mention is here omitted.

It will be remembered that the cutting off on both ends of the cylinder can be rectified by lengthening the eccentric rod, but that this proceeding will render the lead unequal for both strokes. Hence, if we could temporarily lengthen the eccentric rods for the time of the cutting-off and restore their original length for the beginning of each stroke, the cutting-off could be equalized without destroying the equality of the lead. Now, we have seen that the link-motion possesses a feature of this description, and by a judicious choice of certain dimensions we may correct one irregularity by the other. This correction, it will be seen hereafter, is accompanied by an increase of the slip, and we have to choose one of two evils: increased slip, which is attended with greater wear on the link-block, or unequal cut-off, which is attended with a less uniform motion. The former is preferred by most practical engineers.

We may consider the subject of rectification in a somewhat different light. It is known that the cylinder takes more steam on the fore-stroke than it does on the return-stroke, if no rectification is attempted. Remembering, now, that with the link-motion we have the power to change at pleasure the degree of expansion, it is easily seen that a raising of the link on the fore-stroke and a lowering on the return-stroke does precisely what we want. There are two means at our disposal that will accomplish this automatically. We may shift the suspension pin of the link, laterally, towards the inside or outside of the link, as the case may require, and employ the "oscillating movement" of the link for the purpose; or we

may place the hanger in an inclined position, turning the "reciprocating movement" to our use.

Attention should, however, be paid to one circumstance. If the link is suspended by its centre, there are two grades (one forward and one backward grade) at which the suspension pin of the link, as the valve is cutting-off at the return-stroke, will have returned to the same position it occupied at the same period of the fore-stroke, and which we may call, for future reference, the "reciprocal grades." At these grades the cutting-off is effected the moment the imaginary ecentric $i°$ (Fig. 47)

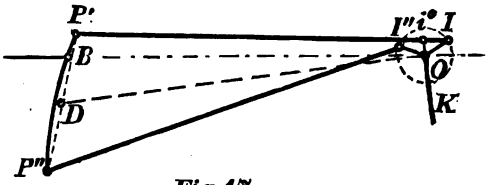

Fig 47.

of the movement of the suspension pin D is at right angles to its imaginary line of operation, OD. The inclination of the hanger can, therefore, not produce the required lowering or raising of the link at either stroke, and we can employ only the oscillating movement of the link for correction.

Since we are restricted to guide the upper end of the hanger in a circular arc, the radius of which is limited to a certain degree, we cannot, after correcting the reciprocal grades, accomplish the perfect rectification of any more than two additional grades, and it is best to choose the full stroke grades for this purpose. The other intermediate grades will then be found so nearly rectified that there will be no need for any further improvement.

The number of grades for which a correction is possible, when the link is suspended on its lower end, is likewise limited to four, and in order to bestow the rectification upon advan-

tageously distributed grades, a careful lateral location of the suspension pin is necessary. If a choice in this respect is excluded, for instance, if one of the pins of the eccentric rods is used for suspension, we must be contented with a less perfect rectification, or as the case may be by bestowing a perfect rectification upon the forward grades only.

The selection of the grades for perfect rectification of the cut-off must be left to the judgment of the designer; it is, however, not necessary to take any of those grades for which the arc of the reversing-lever is notched, as any other intermediate grade is equally serviceable. As a rule, the full stroke grades, and those for cutting-off at half stroke, for both forward and backward rotation, are a fair base for correction, and they have been chosen in the following practical examples.

After these preliminary remarks we may take up the discussion of the way for finding the proper dimension of the suspension mechanism.

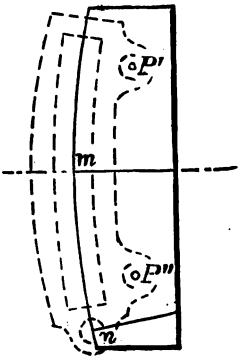

Fig 48.

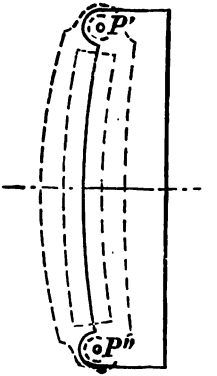

Fig. 49.

The complication and variety of the disturbing influences

render it almost impossible to bring them fully to account without the construction of an actual model, or an equivalent thereof. Such a model can be prepared of stiff drawing-paper or thin wood, and attached, by thumb tacks, to the drawing-sheet on which the suspension lever is to be laid out. It should always be made to full size to insure sufficient precision.

The principal dimensions of the hanger, the link, the eccentric rods, the eccentrics, etc., should be determined to suit the special case, and, as shown in the two problems, the first of which shall be used for exemplification, supposing the connecting rod to be equal 5 cranks, and the hanger $=16''$.

The link may be made as shown in Fig. 48 or 49, and should be cut carefully along the centre-line of the slot. The eccentric rods may be made of two strips of paper and provided with two pin-holes at their ends, at the exact length of the contemplated eccentric rods, and they should be marked "forward" and "backward," or "+" and "—", for contradistinction.

We now draw the centre line of the link-motion (Fig. 50) and

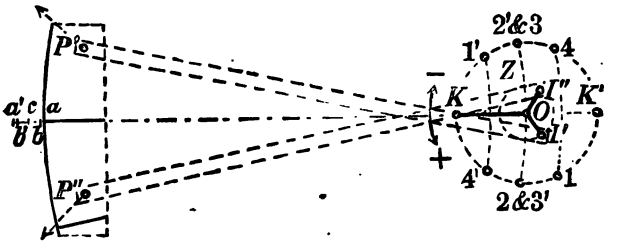

Fig. 50.

sweep from O the orbit Z of eccentrics. Then we construct the crank-angles, at which the cutting-off of the four grades that we selected for the base of correction (which are $\frac{4}{5}$ and $\frac{1}{2}$ of the stroke), takes place, by the proper circular projection.

In Fig. 50 these crank-angles are marked 1, 2, 3, 4, and 1′, 2′.

3′, 4′, the former standing for the fore-strokes and the latter for the return-strokes.

The link can now be attached to suit any crank-angle. by finding, by means of the angles of advance, the corresponding positions of the two eccentrics, and pinning the eccentric rods to those points, care being taken not to pin the "forward rod" to the "backward eccentric." The free ends of the rods may be fastened to the link by thumb tacks, with their points up.

The mode of procedure is now as follows: Attach the link to the first dead centre of the crank K, as shown in Fig. 50 by dotted lines, pull slightly in the direction of the darts, to stretch the paper straight, set the link to mid-gear, and mark with a sharply pointed pencil the point a. Repeat this operation for the second dead centre of the crank, K', and obtain the point a'. Bisect the distance $a\,a'$ in c and regard this point the centre of movement of the link-block. In this way we secure an equal lead as regards fore and return stroke at the mid gear. If the link-block is attached to a rocker, let the arc of the movement of the block pass through the points a and a'. Then mark the positions of the block, b and b', at which the valve will just close the ports by making $cb=cb'$ equal to the outside lap of the valve (duly reduced, of course, when a rocker of unequal leverage is interposed).

Now, suppose the link was to be suspended at its lower end, somewhere in the line n (Figs. 48 and 50). Attach the link successively to all the crank-angles marked before, adjust its curved edge to b or b', accordingly as the crank-angle pertains to the fore or the return stroke, and mark the position of the line n of the link and its termination (or intersection with the link-curve) on the drawing. We thus obtain the eight lines shown in Fig. 51, where they are marked by the same figures as the corresponding crank-angles. They represent, as it were, the positions the link will occupy at the respective periods.

We have now to find the lateral location of the suspension

pin on the line n of the link. To this end we first suppose it to be located in the centre-line of the slot, or, in other words, in the termination of the line n. We sweep accordingly from the end points of the lines 1 and 1′ two arcs of a radius of 16″ (the length of the hanger). Their intersection gives the point G^o from which the hanger should be suspended for the first grade The points of suspension, GG, for the three other grades can be found in the same manner, but it will be seen that no one circle can be drawn through these four points, hence they do not satisfy the required conditions. However, if we move the suspension pin of the link laterally, we can find, after a few attempts, a point, marked D on every one of the lines of Fig. 51 that furnishes points of suspension, g, g, which can be connected by a circular arc. The centre, F, of this arc determines the location of the fulcrum, and its radius the length of the required suspension lever.

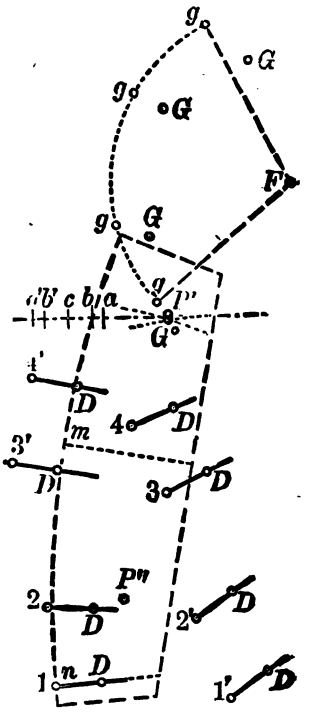

Fig. 51.

The point D can now easily be transferred tothe link, and the designer can accordingly finish the detail drawings.

The above course of procedure will have to suffer a slight modification if the link is to be suspended in its centre-line m

(Fig. 48). The eight different locations of the link must be marked by the centre-line m, and will appear as shown in Fig. 52. It will be observed that the lines 2 and 2′, as well as 3 and 3′, intersect in a point from which the end points of these lines are nearly equidistant, simply because the grades 2 and 3 are in close proximity to the reciprocal grades. By rights we ought to have taken those latter grades instead of 2 and 3, but we obtain practically the same result by averaging the distances of the said points of intersection from the termination of the respective lines and take this distance for the lateral position of the suspension pin D, which then can be located on the eight link-lines. Next we determine the points of suspension, G' and G'', for the grades 1 and 4, and mark through these points the arc of the suspension lever. The latter should be made as long as convenient, in order to have the intermediate grades more nearly corrected.

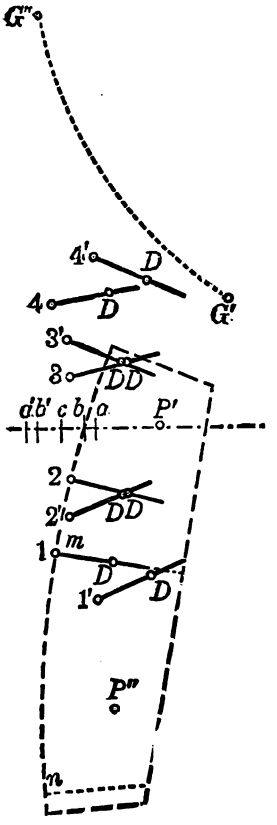

Fig. 52.

When a link-motion is rectified in this manner as regards the cutting-off, it will be found to be nearly so for the cushioning and the release. One of the agencies

employed for the rectification, namely, the inclination of the hanger, will slightly affect the lead, especially at the full-stroke grades, but the difference is so small that it can hardly be noticed. The port-opening, indicated for any crank-angle by the diagram, will also slightly deviate from the actual port-opening of the link-motion, and will generally differ at the corresponding periods of both strokes. The greatest port-opening may thus become smaller than premised.

It must not be overlooked that the rectification is accomplished at the expense of the slip, and it is therefore necessary to sufficiently lengthen the slot of the link for this purpose. The paper model affords the means for finding the extent of this lengthening.

The slip may sometimes become excessive, and it may then be expedient to abandon the perfect correction and compromise between this correction and the slip, by suspending the hanger more nearly in a vertical line than demanded by the above rule. Or the valve can be made with unequal lap for both ports, correcting the cut-off at the expense of an equal lead.

The described manner of attaching the eccentric rods and the link by thumb tacks may involve some inaccuracy. Greater precision may be attained by drawing the link on tracing-paper, and instead of using the paper eccentric rods, sweeping arcs from the respective positions of the eccentrics, and by adjusting the pin-centres of the link to those arcs, the position of the link can be obtained as before.*

* This latter method for finding the position of the link for any given crank-angle has been used in England. Auchincloss, in his treatise on "Link-motion," recommends the use of a template cut of thin wood, to be adjusted to those arcs.

The Gooch Link-Motion.

This link-motion was invented independently of and almost simultaneously with the Stephenson link-motion, and is distinguished from the latter principally in that the link is suspended at a stationary hight, the link-block being moved instead, by the reversing lever, as illustrated in Fig. 53. The

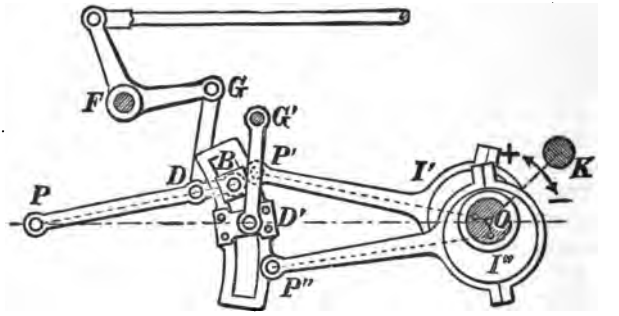

Fig. 53.

connection between the link-block and the valve-stem is effected by the radius rod BP.

To investigate the theory of this link-motion, let us again neglect the influences of the oscillation of the several members, and consider the movement of the pins P' and P'' to be strictly horizontal. The centre line of this link-motion is the line passing through the crank-shaft O and the pin P of the valve stem. The movement of the chord $P'P''$ of the link is now precisely the same as that of the shifting link when in mid-gear. Supposing that I' and I'' (Fig. 54) were the eccentrics, and $P'P''$ the link in its neutral position, we can find the ideal eccentrics J' and J'' of the movements of the pins P' and P'', by moving the real eccentrics I' and I'' through the same angle (β) and in the same direction as the corresponding eccentric rods would have to be moved to make them horizontal, and then

drawing the lines $I'J'$ and $I''J''$ at right angles to OI' and OI''. The horizontal movement of any intermediate point, b, of the chord of the link can again be shown to be equal to the horizontal movement of a corresponding point j of the line $J'J''$, in the same way as before (see Fig. 31); and since in this link-motion the movement of this point is directly transmitted to the valve when the link block is raised to the point B (the movement of the points B and b being practically equivalent,) the point j is the ideal eccentric of the movement of the valve, and the ideal eccentric for any other grade is situated in the straight line $J'J''$, dividing it in the same proportion as the position of the link-block divides the chord of the link. The relation existing between the points b and j can again be established by parallel lines. Draw a line through J' parallel to

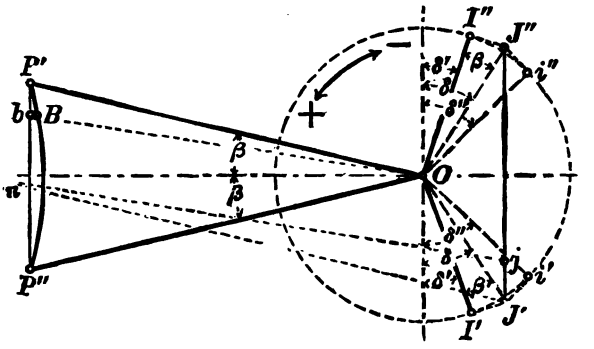

Fig. 54.

$O\,P'$ and extend it to its junction, at n, with the horizontal centre line. The line joining j and n will then be parallel to that joining O and b. The valve diagram, Fig. 55, can now readily be drawn, showing the lead of this link-motion to be equal throughout all grades. In other respects, the feature of

the variableness of the expansion and the exhaust is very much like that of the Stephenson link-motion.

The curvature of the link, having its convex side turned towards the crank-shaft O, should have a radius equal to the length of the rod BP (Fig. 53) as will be seen by setting the link in its neutral position, where a lowering or raising of the link-block must not produce a movement of the valve. The link is mostly suspended, having its centre in the centre line.

If crossed eccentric rods are employed, the eccentrics should have the positions i' and i'' (Fig. 54), when the corresponding eccentrics will again be represented by J' and J'', and the diagram of the ideal eccentrics will be identical with the one for open rods.

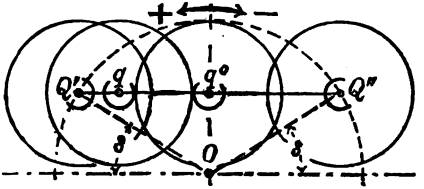

Fig. 55.

To reduce the slip to a minimum both suspension rods GD and $G'D'$ (Fig. 53) should be vertical when in their neutra position. To this end the point G should be guided in an arc equivalent to the arc made by the point D when the link is in its neutral position. The fulcrum F' should, therefore, be vertically above the neutral position of the pin P, at a height equal to the length of the hanger, GD; and the length of the lever $F'G$ should be equal to the length PD. In practice, however, this lever is usually made shorter, and is located to effect a rectification of the cutting off at fore and return strokes.

The advantage of this link-motion over the one treated before consists in the permanency of the lead throughout all grades,

while the increased number of parts, the less length that can be given to the eccentric rods at a given distance of the crank-shaft from the valve-chest, and the greater side pressure that the valve-stem receives from the inclined rod, involving the necessity of a guide for the valve-stem, are doubtless disadvantages.

Problem X.—A Gooch link-motion shall be designed to cut off, when in full gear, at three-quarter stroke, and to cushion for one-twelfth of the stroke. The greatest port-opening is required to be $1\frac{1}{8}''$ with a lead of $\frac{1}{8}''$. The link is $12''$ long and the eccentric rods are $2'\ 1\frac{1}{2}''$ from centre to centre. It is required to draw the valve diagram and to find the throw and position of the eccentrics for both open and crossed rods.

The centres Q' and Q'' (Fig. 55,) for the full gears can again be found as usual, the construction yielding an outside lap $=1''$, an inside lap $=\frac{1}{16}''$ and the travel of the valve $=4\frac{1}{4}''$. To find the required throw and position of the eccentrics we draw the link in its neutral position (see Fig. 54), to find the angle β, carry the same to both sides of the lines OJ' and OJ'' (J' and J' to correspond with the points Q' and Q' of the valve diagram, Fig. 55), and draw the lines $J'I'$, $J'i'$, $J''I''$ and $J''i''$ at right angles to OI' Oi', OI'' Oi'', respectively. The points I' and I'' represent the required eccentrics if open eccentric rods are used, and i' and i'' those for crossed rods. The throw of the eccentrics can be measured by the lines OI', etc., and is found $=2\frac{1}{16}'$. The angular positions of the eccentrics are given either by the angles δ' or δ''. Both eccentric will then have an angle of advance $=\delta$ in relation to their eccentric rods.

The proper points of the link producing certain grades of cut-off can easily be found by the use of the valve diagram.

A rectification of the cut-off of four grades can be accomplished, as in the Stephenson link-motion, through the oscillation of the link and the movement of the hanger GD. The stationary hanger $G'D'$ cannot be made available for this purpose, and should, as a rule, be suspended vertically, in its neutral

position, except if the foward grades are to be favored at the expense of the backward grades; for the rectification of the backward grades will call for exactly the opposite inclination as the rectification of the forward grades. A paper model (see Fig. 56,) can again be used to find the suitable position of the

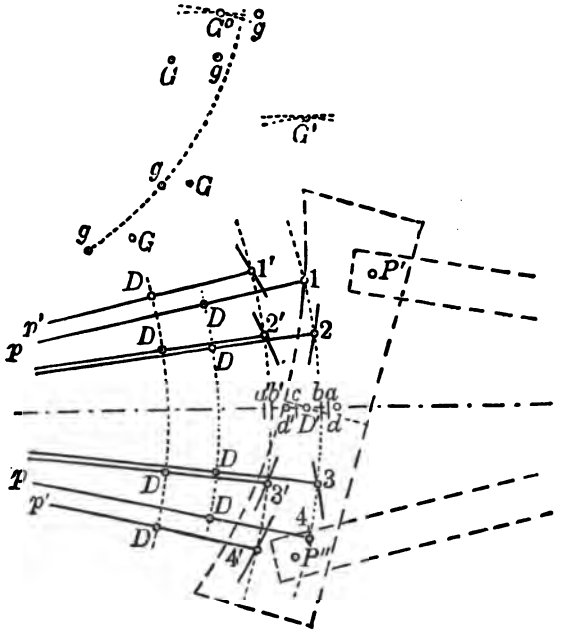

Fig. 56.

stud D and the suspension lever FG. After drawing the orbit of the eccentrics, and the crank-angles for the cutting-off of the four grades, we desire to equalize (compare Fig. 50), we attach

the link to correspond with the dead centres of the crank, mark the points a and a' (Fig. 56), bisect their distance in c and mark the points b and b' representing the position of the link-block (in mid-gear) when the cutting-off takes place. Then we sweep the paths, through b and b', of the link-block when raised or lowered for the said positions of the valve. Next we assume, at pleasure, a point of the link, say D' (Fig. 56) for the suspension pin, and make $bd=b'd'=mD'$, in order to locate the suspension fulcrum G' by sweeping two arcs from d and d', of a radius equal to the length of the hanger $G'D'$. This being done we set the link for the crank-angle 1, adjusting the point D' of the link to the proper distance from G' and mark the intersection 1 of the link-curve with the arc b. We repeat this for the crank-angle 1', marking the point 1' on the line b', and continue with the other crank-angles to obtain the points 2, 2', etc. Draw the corresponding positions of the valve-rod $1p$, $1'p'$, etc. (the points p and p', towards which the two sets of lines converge, are not shown on the drawing), and mark the points D,D, of the suspension pin of that rod. Two arcs swept from the points D,D, of the lines $1p$, and $1'p'$ locate by intersection the point G^0 from which the hanger of the valve-rod should be suspended for having the first grade rectified. The points of suspension G,G,G, for the other grades will be found by the same method.

But, whereas, these four points are not located so that one circular arc can pass through them, they do not suit our purpose. We have, therefore, to change the assumed position of the suspension pin D' of the link laterally, and to repeat the above operation until the suspension points can be connected by a circular arc, which then determines the length and position of the suspension lever.

In the case shown in Fig. 56 the suspension pin D would have to be on the other side of the centre line of the link to obtain this result and the location of the suspension lever would be very awkward. The stud D may, therefore, be located in the

centre line of the link, when the points *gg* will be found and the suspension lever may be made to pass in their close proximity, as indicated by a dotted arc. It will now be noticed that this arc passes exactly through the points of suspension for the backward grades, but is less favorable for the forward grades. Since the reverse would be the more desirable, we may reverse the whole arrangement, *i.e.*, locate the suspension lever below the centre line of the link-motion. Or, if a reduction of the slip is more desirable, the valve may be made with unequal lap when the correction of the cutting-off can be conveniently accomplished with a minimum amount of slip, at the expense of the equality of the lead. Unequal lead is of very little consequence, because the position of the piston for the moment of the opening of the port will be scarcely affected by a slight change of the lead.

Another expedient, sometimes adopted to better equalize the distribution of steam for the forward grades, at the expense of the backward grades, consists in suspending the link so that its centre is below (or above, as the case may require) the centre line of the link-motion.

PART III.

INDEPENDENT CUT-OFF GEARS.

Independent Cut-Off Gears.

As perfect as the link-motions may appear as regards the variableness of the cutting off, they notwithstanding possess one undesirable feature. They admit of a regulation of the *admission* of steam, but the *exhaust* is dependant on this regulation, and the objection existing against the use of an early cut-off by the common slide valve, holds good for link-motions. For this reason the "independent cut-off valves" have been devised, which are combinations of two or more valves that admit a variation of the cut-off independently of the exhaust. They are very extensively in use on larger stationary and marine engines, but their applications to locomotives did not meet with much favor on account of their complication.

The functions of admission and of exhaust may be entirely separated, *i. e.*, distinct valves may be used for each of those functions; or a main-valve similar to that of the common slide

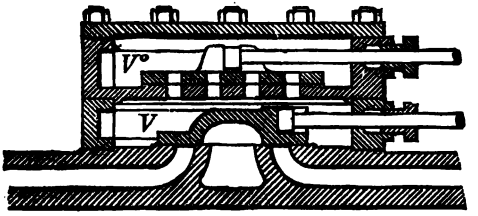

Fig. 57.

can be made to control both admission and exhaust, leaving to the second or cut-off valve the sole duty of closing the admission of the steam at an earlier period than the main-valve does.

The first class includes the disengagement valve gears, which are generally of a more complicated construction, although their theory is very simple. Gears of the second class are less complicated, but their study offers more difficulty, and this class will be considered in the following:

Such valves may be arranged in two different ways. In Fig. 57 a partition divides the interior of the valve-chest in two chambers, in each of which one of the valves are placed, which are moved independently of each other. Both valves are shown in their neutral position. The cut-off valve V^o, is a plate with rectangular perforations or ports coinciding with equal perforations of the partition. A movement of this valve exceeding the width of these ports will interrupt the admission of steam. One single opening in both the valve and the partition valve would answer, but it is preferred to make a number of smaller ports, in order to reduce the travel without reducing the total port opening, for the sake of reducing the loss of power from friction. Such valves are known by the name of "Gridiron valves."

The main valve V is of the same construction as the common slide valve.

The second way is shown in Fig. 58. The cut-off valve V^o is placed directly over the main-valve V, and to make it effective, the main valve is extended on both sides to form two

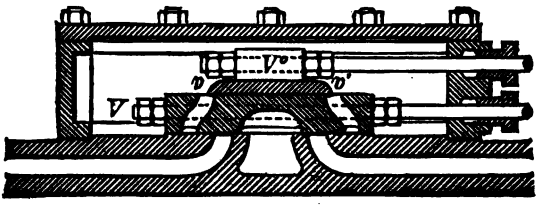

Fig. 58.

steam passages, v and v'. These valves are again shown in their neutral position, to show their relative proportion and their laps; they will, however, rarely attain this position simultaneously when in action.

These valves are usually operated by two separate eccentrics. The action of the main valve V is identical with that of the common slide valve; the cut-off valve V°, in closing the passages of the main valve, will interrupt the admission of steam earlier than the main valve will. The operation of this cut-off valve requires a special study; for the present case differs from our former investigations in that this valve is not sliding over a fixed seat, the seat being formed by the main valve, which has a movement of itself. What we have to consider is not the actual movement of the cut-off valve, but *its movement in relation to that of the main valve.*

Turning our attention to the position of the eccentrics I and I° (the latter being the cut-off eccentric) in relation to the crank $O\ K$ (Fig. 59), we note the movement of the main valve to be represented by the distance $O\ t$, where t is the horizontal projection of the eccentric I. The *absolute* movement of the cut-off valve is the distance of the projection t° of the cut-off eccentric I° from the centre O, but this distance is of little value for our investigation. The *relative* movement of the cut-off valve, *i. e.*, the distance of the centre-line of the cut-off valve from the centre-line of its seat, the main valve, is the distance $t\ t^\circ$, and this distance is the horizontal projection of the line $I\ I^\circ$.

If we now draw the line $O\ J^\circ$ parallel to $I\ I^\circ$, or in other words, if we shift the line $I\ I^\circ$ parallel with itself along the centre line $O\ I$ of the main eccentric to $O\ J^\circ$, it is evident that the projection $O\ t'$ of the line $O\ J^\circ$ must be equal to the projection $t\ t^\circ$ of the line $I\ I^\circ$, which is equal to the relative movement of the cut-off valve; and if we can prove this relation of the line $O\ J^\circ$ to remain unimpaired by a rotation of the

crank, then the point J° must be regarded as the *ideal eccentric of this movement.*

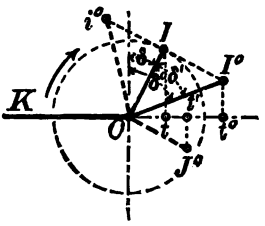

Fig. 59.

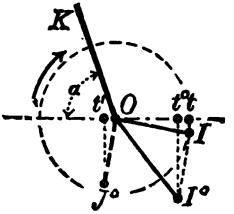

Fig. 60.

To prove this relation, let us view the case after the crank is rotated through the angle a (Fig. 60). The relative movement of the cut-off valve (which, by the way, is negative in Fig. 60), is again represented by the distance $t\,t^\circ$, which is likewise equal to the projection $O\,t'$ of the line $O\,J^\circ$.

The *relative* movement of the cut-off valve is therefore equivalent to the movement of the same valve over a fixed seat when operated by the eccentric J°. This conclusion renders possible the application of the valve diagram, as will be shown hereafter.

The Gotzenbach Cut-Off Gear.

The valves of this gear are like those represented in Fig. 57, each of them being moved by a separate eccentric.

To investigate the action of the cut-off valve, we must consider that the ports are fully opened when the valve is in its neutral position, and that the port opening is reduced by a movement of the cut-off valve, no matter to which side the movement takes place. For the latter reason it is indifferent whether the cut-off eccentric I^o is secured to the shaft in the position shown in the diagram, Fig. 61, or diametrically opposite.

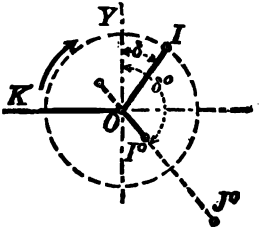

Fig. 61.

The four ported valve (Fig. 57), having the same effect as a valve with only one port of four times the width of the ports shown, and four times its travel, we should consider the ideal eccentric of the movement of the cut-off valve to be in J^o, the distance $O\,J^o$ being equal to four times the throw, $O\,I^o$, of the eccentric I^o. The lap-centres, Q and Q^o, of the valve diagram, Fig. 62, can be constructed in the usual manner by rotating the diagram Fig. 61, through 90° and inverting it, *i. e.*, by carrying the angles of advance, δ and δ^o from the horizontal centre line in

a direction opposing the rotation of the crank-shaft. The cut-off valve is in fact a double valve (closing the ports when moving either way), with a negative lap, on both sides, equal to the collective width of the ports, and the lap circles L^o can accordingly be drawn. The opening of the ports for the crank angle $O K$, for which the collective movement of the valve, *i. e.*

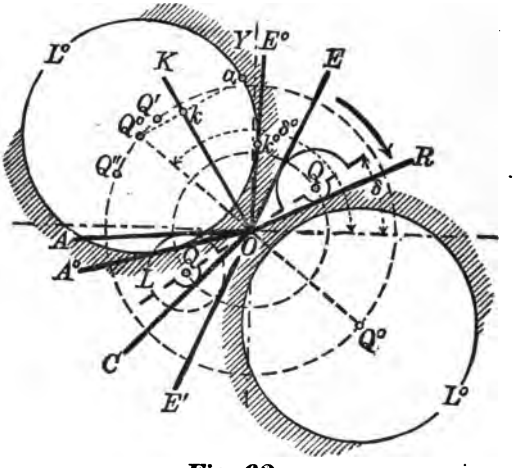

Fig. 62.

the real movement of the valve multiplied by the number of ports, is represented by the line $Q^o k$, is found by deducting this movement from the collective width of the ports (collective negative lap) $Q^o a$, hence the distance, ka, shows the sought total opening of the ports. We can therefore conclude that the cut-off valve will admit steam as long as the crank-line passes through the inside of the lap circle L^o.

The diagram (Fig. 62) can now be completed by adding that of the main valve, when the operation of the double valve will be seen very plainly. The admission of steam begins at OA,

when the main valve opens the port of the cylinder, the cut-off valve having been opened before, at $OA°$. At $OE°$ the cut-off valve closes its ports very sharply, as attested by the comparative length of the line $Ok°$ exceeding even the length OQ or the maximum speed of the main valve, and expansion commences. The exhaust, indicated by the diagram of the main valve, is, of course, not affected by the cut-off valve.

The most effective way of varying the rate of cut-off is a change of the angular advance of the cut-off eccentric, but the diagram will show that this change is limited.

A reduction of the angle $\delta°$ (Fig. 62) will delay the cutting-off, the point $Q°$ being shifted towards Q'. This change will, however, also involve a delay of the opening of the cut-off valve at $OA°$, which should never be carried beyond the line OA where the main valve opens the port.

A hastening of the cut-off can be accomplished by increasing the angular advance $\delta°$ (advancing the cut-off eccentric on the crank-shaft) when the point $Q°$ of the valve diagram will be shifted towards Q'', and the cutting-off will take place earlier than before. This measure must, however, not be carried too far, as the opening of the cut-off valve at $OA°$, might be hastened so much as to take place before the main valve has closed, at OE', the admission at the preceding stroke, and steam would be admitted to the cylinder twice at every stroke. This would evidently be very wasteful, as the steam admitted after the second opening would not return an adequate amount of work.

The range of variation, admissable without trespassing these limits will depend on the outside lap of the main valve; for the lap circle L defines these limits, and hence, by giving the main valve ample lap, the admissable range of cut-off can be made to cover nearly all grades desirable in practice.

The advantages of this valve gear consist in the sharpness of the cutting-off and in the simplicity of its construction; but the inconvenience of having to stop the engine for changing

the cut-off impedes its general adoption. The utilization of the steam is somewhat injured by the circumstance that the lower chamber of the valve chest remains in communication with the cylinder after the cutting off, thus increasing, as it were, the space of the clearance.

Problem XI.—Construct a valve gear of the above description, having a maximum follow of steam of two-thirds of the stroke and a greatest port opening of $1\frac{1}{4}''$, when adjusted to $\frac{1}{16}''$ of lead. The number of the cut-off ports shall be three, and each is to be $1\frac{1}{8}''$ wide. Draw the diagram, determine the highest attainable degree of expansion, and find the amount of cushioning if the inside lap is equal to zero.

We give the main valve as much lap as possible by making it to cut off as early as the required conditions permit, which is at $\frac{2}{3}$ of the stroke. The diagram (Fig. 63) of the main valve

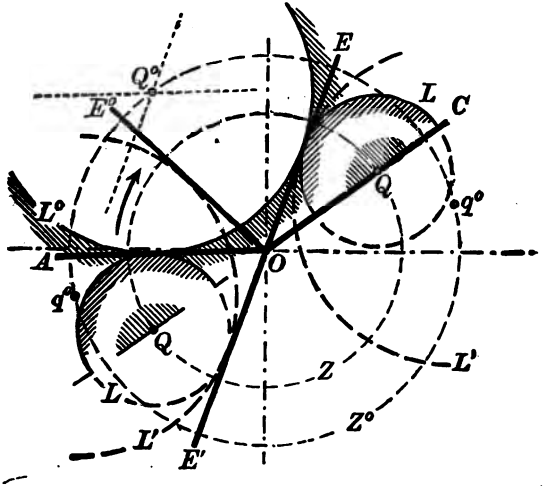

Fig. 63.

can thus be drawn, and will furnish the following dimensions: Lap=$1\frac{5}{8}''$, and travel=$5\frac{3}{4}''$. The compression is found to last for $\frac{1}{10}$ of the stroke. The cut-off valve should now admit steam between the crank-lines OA and OE; and since we know the radius of the lap circle $L°$ of the cut-off valve to be $=3\frac{3}{8}''$, we can find its centre by two lines drawn parallel to OA and OE at a distance each of $3\frac{3}{8}''$, and can draw the same tangential to $O\,A$ and $O\,E$. One third of the distance $O\,Q°$ is the required throw of the cut-off eccentric, which is found to be=$1\frac{3}{8}''$. An advance of the cut-off eccentric for the purpose of hastening the cut-off, can only be carried so far that the lap circle L' becomes tangential to the cut-off line $O\,E'$ of the main valve, as indicated. $O\,E°$ will accordingly be the crank angle for the earliest cut-off, and is found to correspond with $\frac{1}{8}$ of the stroke. The angularities of the eccentric rods might, however, admit a slight opening of the cut-off valve before the main valve closes, at $O\,E'$, and therefore should the eccentric never be advanced quite as far.

This cut-off gear is frequently used on marine engines in connection with link motions operating the main valve. The change of expansion is, however, accomplished by a change of the *travel* of the cut-off valve by means of a separate link worked by the cut off eccentric, on a fixed fulcrum. An investigation will show that by this plan the range of expansion is very limined indeed, the centre $Q°$ of the valve diagram (Fig. 62) moving in or out on the line $OQ°$ or on a curve* not much deviating from this line. By combining this cut-off valve diagram with the diagram of the main valve for *backward* rotation, it will appear that this cut-off valve is unserviceable altogether, admitting steam twice at each stroke and withholding it at the intervening period. The cut-off valve must

*The character of this curve will be more closely investigated in a subsequent chapter, where its form is of more consequence.

therefore be thrown out of gear when the engine is backing, by shifting the cut-off link-block close up to the fulcrum of the cut-off link, to keep the valve open entirely.

The unequality of the cut off, owing to the oscillation of the connecting rod, can be rectified by giving the cut-off valve unequal lap, *i. e.*, by lengthing or shortening the valve stem, according as the cut-off eccentric is located at 1° (Fig. 61) or diametrically opposite. This correction can however be made to suit only one angle of the connecting rod, and it is well to choose the angle corresponding to $\frac{1}{3}$ of the stroke of the piston, when the same correction will also cover the cutting off at $\frac{2}{3}$ of the stroke, the angle of the connecting rod being the same.

In practice, the cut-off valve can easily be adjusted on the work. But if it is desired to know the amount of adjustment before hand, it can be found by the diagram as indicated in Fig. 64.

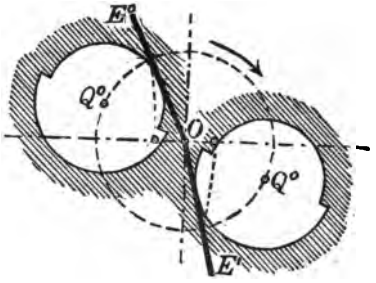

***Fig.* 64.**

The Meyer Cut-Off Gear.

The valves of this gear are substantially of the type represented in Fig. 58 and are moved by two separate eccentrics, located on the shaft as indicated in Fig. 59. The cut-off valve, however, differs somewhat, in being made of two parts (see Fig. 65) that are attached to the valve-stem by means of a right and a left-handed thread. The valve-stem is joined to the eccentric rod by a swivel, and can be rotated by a hand wheel or other suitable mechanism, whereby the distance of both cut-off plates can be adjusted, even while the engine is running. The lap of this valve can thus be changed at will.

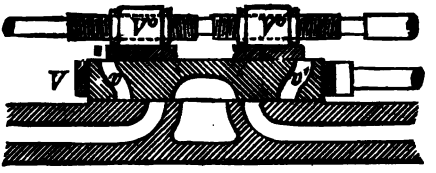

Fig. 65.

For converting the diagram of eccentrics (Fig. 59) into the valve diagram (Fig. 66) beginners are advised to make use of tracing paper to locate the points Q and Q' relating to the main eccentric I, and $Q°$ and Q'' relating to ideal eccentric $J°$. The relation existing between the diagrams Fig. 66 and Fig. 59 involves the following inferences. The distance $OQ°$ is equal to one-half the *relative* travel, and the distance $Q'Q°$ equals one-half the *absolute* travel of the cut-off valve; and, if we draw the horizontal line h through the point Q', the angle

$hQ'Q^\circ$ and $OQ'Q^\circ$ will equal the angles δ° and δ' of Fig. 59, respectively, these angles appearing, in the valve diagram, in a direction opposite to that in the diagram of eccentrics. *The line $Q'Q^\circ$ represents manifestly the cut-off eccentric in the same sense as the line OQ represents the main eccentric.*

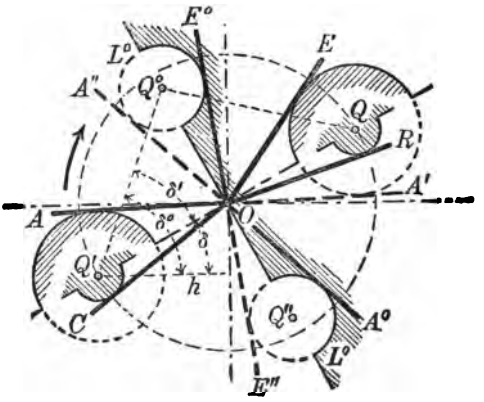

Fig. 66.

By sweeping the proper lap circles from the points Q and Q', the diagram for the main valve can be completed, and that of the cut-off valve can be drawn to suit the lap of the cut-off valve for the time being. Assuming, at first, a permanent negative lap, we draw the lap circles L°, as shown, and find the passage v (Fig. 65) to be opened while the crank moves from A° to E°, but as the main valve opens the port of the cylinder only after the crank passes the line A, the admission of steam will take place between A and E°. The passage v' of the main valve will remain open between A'' and E'', and hence the steam will be admitted, at the return stroke from A' to E''.

A change of the lap of the cut-off valve requires the drawing of other lap circles from the point Q^o, and the diagram will appear as shown in Fig. 67, where four grades of expansion are represented, two with positive and two with negative lap. The lower mate of Q^o is omitted, being of little interest since this diagram shows the action on but one side of the piston.

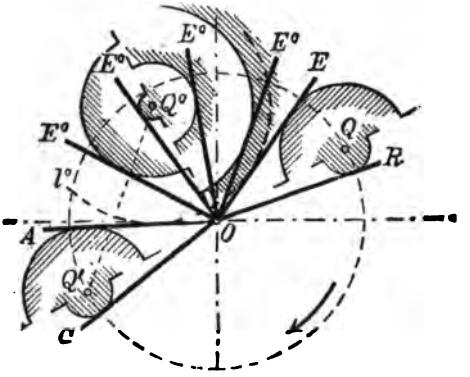

Fig. 67.

A glance at this figure now shows that the range of expansion includes all desirable grades, provided the cut-off valve admits of sufficient adjustment; and also the sharpness of the cutting off shows a fair figure. The latter is greatest when the lap equals zero, and becomes less as the lap is either increased or reduced from this poin

However, the margin of adjustment of the cut-off valve is practically limited, as will be shown in the following problem, and this fact puts a limit either to the range of expansion or to the rapidity of the cutting off; the latter case involves a reduction of the relative travel of the cut-off valve, or, in

other words, a reduction of the distance of the cut-off eccentric l° (Fig. 59) from the main eccentric I.

In order that the rapidity of the cutting off should suffer the least reduction at the change of grades, it is best to so locate the cut-off eccentric that this rapidity is equal at the two extremest grades, and this is accomplished if the positive lap of the earliest cut-off equals the negative lap of the latest cut-off desired. Therefore, if it should be required to make the cut-off adjustable between the crank-angles A and E, (Fig. 67) the lap-circle l° would stand for both extreme grades, and this circle can be tangential to both the lines OA and OE only, if the line OQ° is at right angles with OQ, and this condition requires the line II° (Fig. 59) to form right angles with the line OI. Such a full range of expansion is, however, not always required, and hence this rule is not a general one.

The question may be raised if this gear will ever admit the steam twice during one stroke, *i.e.*, if the cut off valve will ever re-open before the main valve closes the steam port. Such an event would be indicated by the diagram if the rotating crank line would meet the line OA° (Fig. 66) before passing the line OE, and this state can only exist if both the negative lap of the cut-off valve and the angle of advance of the cut-off eccentric are excessive, and can, therefore, be avoided.

This valve gear is susceptible of quite a number of modifications. The duty of cutting off may be imposed upon the inside edges of the cut-off plates, by their being farther apart while the upper terminals of the steam passages of the main valve are closer together. The difference between this arrangement and the one considered before consists in that the relative movement should be exactly the opposite to produce the same result. We should here remember that the diagram furnishes only the angular position of the eccentric, leaving a choice between two diametrically opposite points, between which the decision is to be made, as explained in a preceding chapter. The same question is now at issue in relation to the ideal

eccentric of the relative movement of the cut-off valve, represented in this instance, is found to require a position diametrically opposite to the point J°, Fig. 59, and the real eccentric should, consequently be at i°.

The same conclusion may be reached in a somewhat different way. If we consider that the points Q and Q°, that were obtained directly from the inverted tracing, always mark the passage of the valve from *positive to negative* movement, it will be understood why the lap circles drawn from those points as centres always marked the closure of the admission at the fore-stroke. But in the last case the cut-off is effected by a *positive* movement, and if a result identical to that represented by Fig. 66 is desired, the point Q° should represent the passage from *negative to positive* movement, and hence its mate, Q'', is the point that should be obtained from the tracing, and it is the line $Q'Q''$, instead of $Q'Q^{\circ}$, that now represents the real cut-off eccentric. Of course, the line QQ° can be substituted for $Q'Q''$, both lines being equal and parallel.

The described mechanism for changing the lap of the cut-off valve is not readily accessible for inspection, and being exposed to a high temperature is apt to wear out in a short time. This gave rise to the invention of other modes of changing this lap. One of these is shown in Fig. 68. The eccentric rod IP is joined,

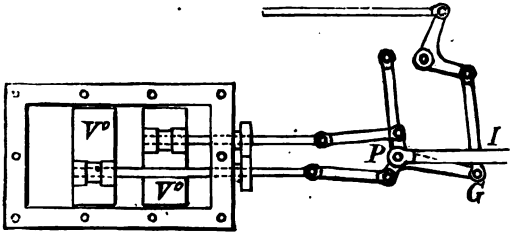

Fig. 68.

other wo
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which,
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are operated. A raising or lowering of
e will change the relative position of the
bjectionable feature of this arrangement
of forces, the valve stems being out of
plates.
n is known as the "Rider Cut-off."* The passages of the main valve have an inclined position, converging towards one side. The ends of the cut-off valve, whose seat on the main valve is cylindrical, are also inclined to conform with the passages of the main valve. A partial rotation of the cut-off valve, by its stem, will bring its wider or narrower side over the passages, and thus change the lap.

There are quite a number of other devices for the same purpose, but their descriptions would occupy too much space.

Problem XII.—It is required to construct a Meyer valve gear for an engine of 20″ stroke, the admission of steam to range between 1″ and 16″ follow. The rapidity of the cutting off at both extreme grades is to be equal to one-half of that of the main valve at the moment it closes the steam-port. The width of the steam-port is $\frac{7}{8}$″, and the greatest port-opening is to be equal to this width. The lead is to be $\frac{1}{16}$″, the release is to take place 1″ before the end of the stroke, and the cushioning is to begin as early as the case will admit.

To solve this problem we first draw the diagram for the main valve with a greatest port opening=$\frac{7}{8}$″, lead=$\frac{1}{16}$, and release=1″ (see Fig. 69). In order to comply with the condition mentioned last, the main eccentric should have as much angular advance as possible, hence we make it to cut off at 16″. Next we draw the crank angle $OE°$ for 1″ of the stroke, which is the earliest point of cut-off required. The centre $Q°$ should be located in a line bisecting the angle $E°OE$, and to get its exact location we must consider that the line Ok (k being the

*Patented.

point of contact of the line OE with the lap circle L) represents the rapidity of the cutting off by the main valve, and in order to comply with the required condition, the point k°, bisecting the distance Ok, should be the point of contact of the line OE, and the greatest lap-circle of the cut-off diagram. We find accordingly the point Q° by drawing a line from k° at right angles to OE. The sought throw of the cut-off eccentric is equal to the distance $Q'Q^\circ$, and the angle formed by both eccentrics is equal to the angle $Q^\circ Q' O$ of the diagram; (or if we choose to cut off with the inside edges of the cut-off plates, the line QQ° and the angle $Q^\circ QO$ are to be taken instead, and the eccentric located like the point i° of Fig. 59).

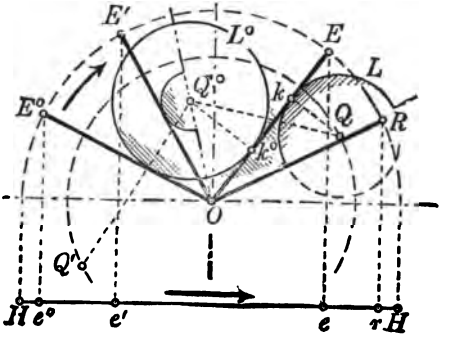

Fig. 69.

By the diagram we find, for the main valve, a travel of 3″ full, a lap $\frac{5}{8}$″ full, the inside lap is equal $\frac{1}{16}$″, and for the cut-off eccentric we obtain a throw, $Q'Q^\circ = 2$″ full. The greatest laps, both positive and negative, of the cut-off valve are $\frac{13}{16}$″ full, and the relative travel of the cut-off valve over the main valve $= 2\,OQ^\circ = 2\frac{1}{8}$″ full.

Since the change of lap must be accomplished on both cut-off

plates, their total adjustment must amount to four times the greatest lap, which is more than $3\frac{1}{4}''$. Attention must also be paid to prevent the *inner edges* of the cut-off plates from opening the passages of the main valve before the latter closes the steam ports of the cylinder. The danger for this occurrence is greatest when the cut-off plates are farthest apart, having a lap$=1\frac{3}{16}''$, and since the main valve closes the port at E when the relative movement of the cut-off valve is equal to $Q°k°=1\frac{3}{16}''$ full, the apprehended opening can be prevented by an overlapping of $1\frac{3}{16}''$ plus $\frac{1}{4}''$ for seal. Hence the width of each cut-off plate must be made equal to this figure ($1\frac{1}{16}''$) plus the width of the passage of the main valve, $\frac{7}{8}''$ plus the lap, $1\frac{3}{16}''$ full$=3\frac{3}{4}''$ full, and their dimension, out to out, when spread apart, will be about $8\frac{7}{8}''$; and since farther the absolute travel of this valve is $4''$ full, the valve chest is required to have a length, inside clear, of about $13\frac{1}{2}''$, allowing about $\frac{1}{4}''$ clearance on each side.

By altering the diagram, with a view of doubling the rapidity of the cutting off, the valve chest will be found to require a clear inside length of about $21\frac{1}{2}''$, and the valve and the cut-off plates would assume rather impracticable dimensions.

If it is desired to know the relative position of the cut-off plates, *i.e.*, their laps, for any grade, say when cutting off at quarter stroke, we simply draw the corresponding crank angle, OE', and from $Q°$, tangential to this line, the lap circle, the radius of which is the sought lap which in this instance is positive. If the crank angle of the given grade were beyond $OQ°$ the indicated lap would, of course, be negative.

When used in connection with a link-motion, the relative movement of the cut-off valve changes with the grade of the link. While the grade of the link is changed, the point $Q°$, Fig. 70 (compare with Fig. 67) will move exactly as the point Q' does, because the line $Q'Q°$ represents the cut-off eccentric,

which remains undisturbed as the link is reversed. The link is, however, generally used at the full stroke grades only, and the diagram, Fig. 70, shows these two grades, capital letters standing for the forward and small letters for the backward rotation. We might have considered the diagram of eccentrics for both grades separately, when we would have obtained the same diagram. Now we see that the range of the cut-off for the backward rotation is limited, the latest cut-off being at $OE°$. This condition can be improved by advancing the cut-off eccentric more or less; and the relation between the forward and backward grades will be entirely equalized if the angle of advance of the cut-off eccentric is made=90° when the points $Q°$ and $q°$ will be removed to Q'' and q'', vertically above Q' and q' at a distance equal to the throw of the cut-off eccentric. It will then be necessary that the greatest negative lap, for the latest cut-off, should exceed the positive lap for the earliest cut-off, as will be seen by drawing the respective lap circles.,

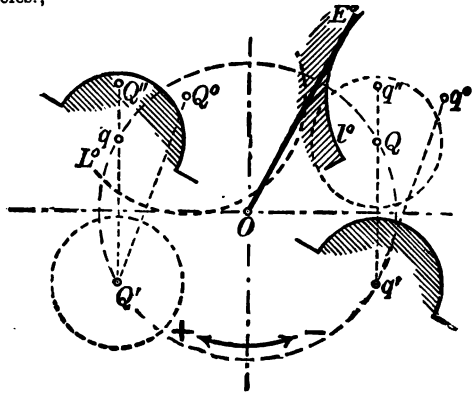

Fig. 70.

A rectification of the cut off can be accomplished in a similar way as with the Gonzenbach valve, and this discussion need not be repeated again. However, with the mechanism shown in Fig. 68, the rectification can be made to cover more than two grades by making the yoke out of straight, as shown in the cut. The effect of this expedient is equivalent to a lengthening of the valve stem as the point of cut-off approaches half-stroke, and *vice versa*.

The Link Expansion Gear.

A change of the "lap" of the cut-off valve is not the only means for varying the expansion, the present valve-gear accomplishing the object by a variation of both the travel and the angular advance.

The main eccentric I (Fig. 71) is connected with the main-valve in the usual way. The cut-off eccentric imparts to the link $P^{o}F$ a rocking motion on its fulcrum F. The link block B is directly attached to the stem of the cut-off valve, the latter being as shown in Fig. 58. A change of the cut-off is effected by a raising or lowering of the fulcrum F, by means of a screw or any other suitable mechanism.

When the link is raised so much that the pin I^{o} is in line with the valve-stem, the cut-off valve will be moved directly

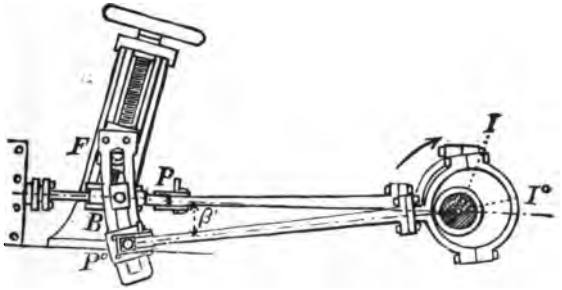

Fig. 71.

by the eccentric I^{o} (see Fig. 72). A lowering of the link implies a double change; first: the angular advance is vari-

by reason of the angular change of the eccentric rod $I^{o}P^{o}$ (Fig. 71.) and second: the travel is altered, the link-block being nearer to the fulcrum of the link. The angular advance is increased by the angle $BOP^{o}=\beta'$, and the travel is reduced in the ratio of BF to $P^{o}F$. We can accordingly find the ideal eccentric i' (Fig. 72) of the corresponding movement of the valve. A raising of the link will bring the ideal eccentric to i''.

For different elevations of the link the ideal eccentrics will again form a curve resembling the one found for the shifting-link, and by starting with similar assumptions this curve will again be a circular arc. In one point, however, there exists a marked difference. The curve of Fig. 72, if extended, passes through the centre O; for if the fulcrum F of the link

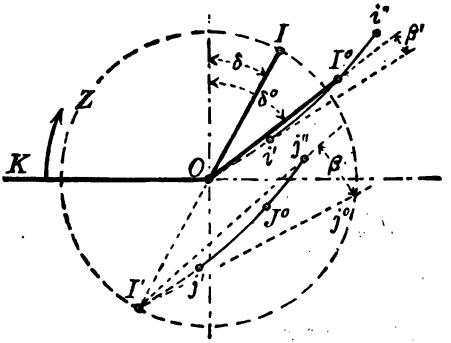

Fig. 72.

(Fig. 71) could be lowered to be in line with the valve-stem, the movement of the valve would be zero.

This curve now relates to the *absolute* movement of the cut-off valve. To obtain the ideal eccentrics of the *relative* movement of the same we have to shift this curve in the direction

of, and through a distance equal to, the line IO, when we obtain the arc $j'\circ^{\circ}j''$, the extension of which passes through the point I' diametrically opposite the eccentric I.

The greatest absolute travel of the cut-off valve is equal to twice the length of the chord $I'j''=Ci''$ of the *locus*, and if we draw the line $I'j^{\circ}$ tangential, in I', to the *locus*, this line will form, with the chord $I'j''$, an angle, β, that is equal to the total angular change of the cut-off eccentric rod, supposing that the slot of the link was extended so far that the fulcrum of the link can be lowered to be in line with the valve-stem. This will be clear when we consider that the vertical inclination of the line $I'j^{\circ}$ represents the angular advance as the link is lowered, while $I'j''$ represents the same as the link is raised all the way.

The valve diagram can now be obtained in the usual manner, and, assuming a negative lap, as shown in Fig. 58, the lap-circles can be drawn, when the diagram, Fig. 73, will give

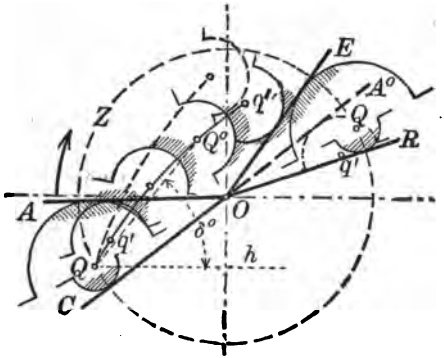

Fig. 73.

us all the information required. We see that the range of variation extends from OA to OE, but we also see that th

rapidity of the cutting off is at the lowest figure at the middle grades that are likely to be used the most. It is true, the rapidity of the cutting off for those grades can be somewhat augmented, by an increase of the angle c° (compare with Fig. 72), whereby the locus will be changed to a position indicated by a dotted arc, but this change is attended by a reduction of the range of the cut-off, this measure materially affecting the limit of the latest cut-off.

A greater curvature of the locus, attainable by lengthening the link, or shortening the cut-off eccentric rod $I^\circ P^\circ$, Fig. 70, or both, will favor the sharpness of the cutting off at the middle grades.

In the case of the fulcrum F (Fig. 71) being on the lower end of the link (the engine running over) the curvature of the locus will be the reverse, and the valve-gear would give poor satisfaction, owing to the reduced travel of the middle grades. For this reason the fulcrum should be on that end of the link *towards* which the crank is rotating, in order to effect an *increase* of the angular advance of the cut-off eccentric simultaneously with a *reduction* of the travel of the valve, and *vice versa*.

The proper curvature of the link may be found by setting both the eccentric rod $I^\circ P^\circ$ (Fig. 71) and the link-block in their neutral positions, when a raising and lowering of the link should not produce any movement of the valve. One glance will show that the curve of the link should then be an arc from the centre O, provided the fulcrum F be moved in the same arc. It is, however, more convenient to guide the fulcrum in a straight line, and in this case the curve may be determined by means of a full sized model; (made of stiff paper or any other suitable material) consisting of the eccentric rod and the link. One end of the eccentric rod is pinned to the centre O of the crank-shaft (on the drawing sheet) and the other end is attached to the link. A pencil point is held

directly over the neutral position of the link-block, and the link is then drawn through under it, by moving the fulcrum of the link in the line representing the contemplated guide. The pencil will thus delineate the required curve, which, though, will be found not to be a circular arc, and it can only be approximated in the practical execution. This curve will be more nearly circular if the fulcrum F is guided in an inclined line, conforming as near as possible to an arc, drawn from the centre O of the crank-shaft.

The lap of the cut-off valve is a factor, which, if not properly determined, may have a prejudicial influence upon the rapidity of the cut-off already limited by the small relative travel. The relative velocity of the cut-off valve attaining its highest rate when the cut-off valve is in its neutral position in relation to its seat, a valve with no lap will effect the cutting off at this period. But as the rapidity of the closure depends not only on the velocity of the valve at the moment of the cutting off, but also on that preceding this moment, it is clear that a negative lap will give better results. The time for the closure of the port will obviously be shortest if the valve has a negative lap equal to one-half the width of the port; for the velocity of the valve is then greatest when the port is half closed. But it must not be lost sight of that in this case the velocity is falling off towards the moment of the cutting off, and the negative lap should never be much greater than one-quarter of the relative travel; for else this reduction is greater than desirable. Hence the negative lap should not exceed one-half the width of the passage of the main-valve, nor should it be much in excess of one-half the shortest distance of O from the curve $q'Q°q''$ (Fig. 73).

But there is still another factor limiting the admissible extent of the negative lap; for negative lap may render possible a second opening of the ports of the main-valve before the latter closes the steam-ports of the cylinder. A brief consider-

ation will, however, show that such an occurrence will be prevented if the negative lap of the cut-off valve is less than the outside lap of the main-valve; for at the grade of the earliest cut-off the passage of the main-valve will be opened at the point $OA°$, which is found by sweeping the lap-circle from the mate of the point q'. This line can evidently not be in advance of OE unless the negative lap of the cut-off valve exceeds the outside lap of the main-valve.

The valve is sometimes made of two plates rigidly fastened together by a yoke, or of a plate with a rectangular opening, the inside edges performing the cutting off. The cut-off eccentric is then placed behind, not ahead, of the main eccentric. The demonstration relating to Figs. 72 and 73 can be applied to Fig. 74, but instead of the points $q'Q°q''$, their upper mates are used in Fig. 74 as centres of the lap-circles showing the cut-off, for a reason explained in the foregoing

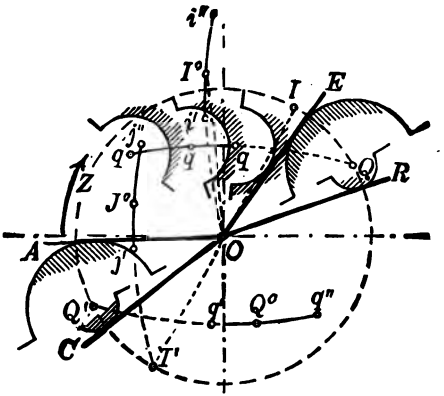

Fig. 74.

chapter. The curve of ideal eccentrics has been determined in Fig. 74 by supposing the fulcrum to be on that side of the

link *from* which the crank is rotating, in order that an *increase* of the angular advance be attended by an *increase* of the travel. The peculiarities of the valve-gear are now considerably changed, and a very early cut off is not attainable except at the expense of the rapidity of the cutting off of the later grades. An improvement can be effected by using either no lap at all or even positive lap, on the cut-off valve. For although this measure is prejudical to a free admission of steam, as it reduces the port opening of the cut-off valve for intermediate grades, the adoption of positive lap permits a change of the angular advance of the cut-off eccentric to favor this shortcoming. The most advantageous lap can easily be found by a few trials, as will be shown hereafter.

This valve-gear admits of various modifications. The fulcrum of the cut off link may be a fixed point, and the cut-off eccentric rod may be attached to the link-block, while the cut-off valve-stem is worked from a fixed point of the link; or the eccentric rod may be joined to a fixed point of the link, and the valve stem may be worked from the link-block by a radius-rod in the manner of the Gooch link motion. In the former case a change of the grade involves both an angular change of the eccentric rod and a change of the absolute travel of the valve; in the latter case a change of the travel only is attainable. In the first case the ideal eccentrics will again form a curve, which, however, will not be a circular arc, and should, therefore, be constructed from the angle of advance, and the absolute travels of the cut-off valve due to various positions of the link-block. In the second case the ideal eccentrics will form a straight line, to the detriment of the sharpness of the cutting off at intermediate grades.

Problem XIII—A valve gear of the description shown in Fig. 71 is to be made having a range of cut-off from $\frac{1}{16}$ to $\frac{8}{16}$ of the stroke. The greatest port-opening of the main-valve

is to be $1\frac{1}{4}''$, the lead $\frac{1}{8}''$, the inside lap=0. The cut-off valve is to be made in form of a frame to cut off with the inside edges, and its greatest absolute travel is not to exceed double the travel of the main valve. The cut-off eccentric-rod, which is 36″ long, is joined to the link at a point 9″ from the fulcrum. The virtual length of the link (the greatest distance of the link block from the fulcrum) is 12″. The throws and positions of both eccentrics and the laps of both valves shall be determined, if it is required that at the grade of the least relative movement of the cut-off valve the time for closing the last half inch of the port of the main valve shall be as short—*i. e.*, that the relative velocity of the cut-off valve $\frac{1}{4}''$ before the closure shall be as great—as possible.

The diagram of the main-valve, cutting off at $\frac{8}{10}$ of the stroke and giving $1\frac{1}{4}''$ port-opening and $\frac{1}{8}''$ lead, is found first, as usually, and the next thing is to determine the locus for the cut-off (compare with Fig. 74). If we first assume the slot of the link extended to the fulcrum, the curve will start at the centre Q (Fig. 75). Since the greatest absolute travel of the cut-off valve is required to equal twice that of the main-valve, we sweep from Q, indefinitely, the arc s with a radius equal to twice the distance OQ. In this arc the locus will terminate. We also determine the crank-angle $QE°$ for the earliest cut-off, and assume, for a start, the lap of the cut-off valve equal to zero, in which case the point q'', the intersection of the lines s and $OE°$, will the be outer end of the locus. Next we bisect the distance $q''Q$ perpendicularly by a right line, in which the centre M of the arc must lie. Thereupon we find the total angle of variation, β, of the eccentric rod by making $fO=bO=36''$, (to reduced scale) and $fb=12''$ and then by copying $q''Qq°=\beta$, we obtain $Qq°$, which must be a tangent, in Q, of the locus. The centre M can therefore be finally located by the line QM, forming right angles with the line $Qq°$, and the locus $q''Q$ can be drawn.

By drawing a line from M through O, we find the point q'

of this arc, representing the least relative travel of the cut-off valve, and sweeping, from q', a circle of a radius of $\frac{1}{4}''$, we obtain the crank angle OE', at which the opening of the steam passage is $\frac{1}{4}''$, and Ok' represents the velocity. After assuming different laps for the cut-off valve and repeating the foregoing proceeding, we can select from the various attempts

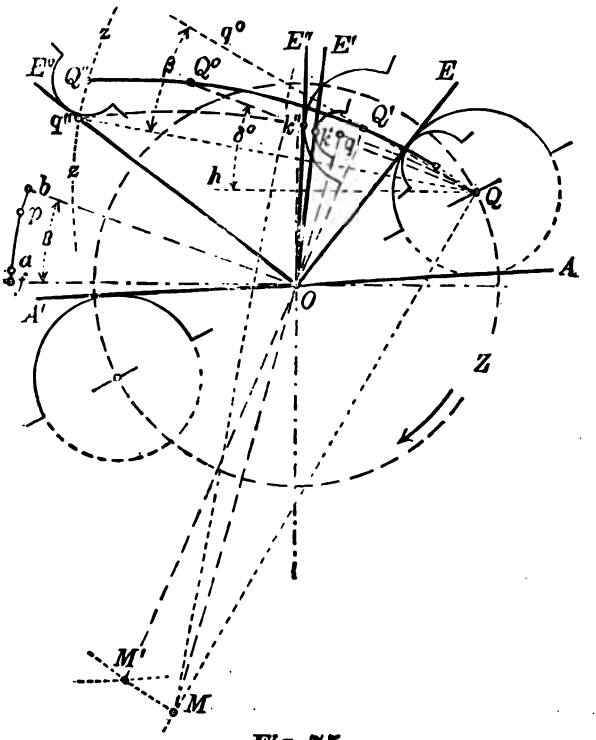

Fig. 75.

that one which gives the greatest velocity. If we assume, for instance, a positive lap of $\frac{3}{8}''$, it is obvious that Q'' (Fig. 7c) of the new locus, must be $\frac{3}{8}''$ from the line $OE°$. The new centre M' of the corresponding locus is found by the intersection of indefinite arcs of equal radii swept from Q (through M), and from Q'. The centre Q' of the least relative travel of the new locus, can again be found by a line from M' through O, and a circle of a radius of $\frac{5}{8}''$ (lap plus $\frac{1}{4}''$) gives us the sought velocity, Ok'', by a tangent from O, which it will be seen, is greater than the result of the first attempt. A further increase or reduction of the lap of the cut-off valve, will be found to furnish less favorable results whence the assumption of the last attempt should be adopted.

The angular advance and the throw of the cut-off eccentric remain yet to be determined. To this end we locate the point p of the link where the eccentric rod is attached, and by dividing the locus $Q''Q$ in the same proportion as this point divides the link, we find the point $Q°$ representing the action of the valve while the eccentric rod is in line with the link-block. Hence the distance $QQ°=3\frac{3}{16}''$ is the throw, and the angle $\delta°$ the angular advance of the cut-off valve. The latter is negative, showing that the cut-off eccentric is located somewhere like $i°$ of Fig. 59. The throw of the main eccentric is $2\frac{1}{8}''$, the lap of the main valve$=\frac{7}{8}''$, and that of the cut-off valve$=\frac{3}{8}''$.

In proportioning the cut-off valve we must observe that the plates are made long enough to prevent the outside edges from opening the passages of the main valve before the latter has closed the respective port of the cylinder. The relative movement of the cut-off valve (for the grade Q'') at the moment the main valve closes the steam port, at E, is equal to the distance of Q'' from the line OE, which is very nearly equal to the greatest relative movement, $OQ''=3\frac{1}{8}''$, the cut-off valve ever attains, hence the outer lap of the cut-off plate should

qual $3\frac{1}{8}''$, with an addition of at least $\frac{1}{4}''$ to secure a seal. To keep the inside edge of the *wrong* plate from covering the passage of the main valve after the latter begins to admit steam, we measure the relative movement of the cut-off valve for the crank-angle of admission, OA', which is the distance of Q'' from this line,$=2\frac{3}{8}''$ about, and see that the inside edge of the *wrong* plate, in its neutral position, will be at least this distance from the passage; and as the lap of the cut-off plates is $\frac{3}{8}''$, their clear distance should be at least $2''$. In the present case, however, this distance will have to be greater, for other reasons, and there is no fear, therefore, of one plate interfering with the duty of the other.

The diagram shows very fair figures as regards the rapidity of the cutting-off, owing to the limited range of expansion. This valve-gear would, in fact, not answer when adjusted by a governor, since the shortest attainable admission, $\frac{1}{10}$ of the stroke, would make the engine run away when without or with but a light load.

Supposing the engine was running over, the cut-off link should have the fulcrum on its lower end, since the cut-off plates are cutting off by their inside edges. The highest position of the link is found by sweeping from the proper point of the locus a lap circle of $\frac{3}{8}''$ radius, tangential to the line OE, the position of the centre of which determines the lowest point, a, of the link, which is found to be so near to the fulcrum that the latter should be secured to the link by a bridge.

This valve gear can be made serviceable in connection with a link-motion only by making the valve as in the last problem, of two plates cutting off with their "inside edges," since otherwise the cut-off valve, negative lap notwithstanding, would invariably cut off at a very early point of the stroke when the link of the main valve is reversed, the locus assuming the position $Q'Q°$, in Fig. 76, while if the valve is made

as said, the locus will be in Qq^o, showing that the cut-off valve even with positive lap, will not cut off, at any grade, before the main-valve closes the steam-ports of the cylinder. For backward motion, then, we have to avail ourselves of the link-motion of the main-valve when we desire to vary the cut-off, and this style of valves can therefore be used to advantage only on engines which are mostly running forward, as is the case with marine engines.

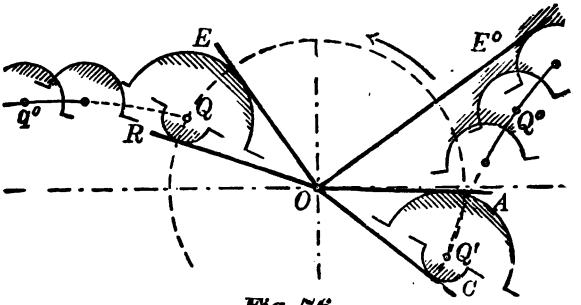

Fig. 76.

A few words on the correction of the unequalities of the fore and return-stroke may be added in conclusion. This valve-gear offers facilities for correcting the cut-off for every grade, as the necessary lengthening of the valve stem can be accomplished by a deviation of the curve of the link from the form defined by the foregoing discussion, and since every grade corresponds to a different point of the link, the precision of a correction throughout the whole range is only dependent on the accuracy of workmanship. The finding of the proper curve of the link will offer no difficulty, if the former discussions on the subject are duly considered.

The Bilgram Cut-off Gear.*

The described gears, adaptable to valves arranged as shown in Fig. 58, have been shown to cut off rather sluggishly—almost throughout with less speed than the main valve alone would—owing either to their very nature or to the practical limitation of certain proportions. The present valve-gear is designed to overcome this difficulty.

Both valves (see Fig. 58) are operated by one single eccentric *I*, Fig. 77; the main valve directly by the eccentric rod,

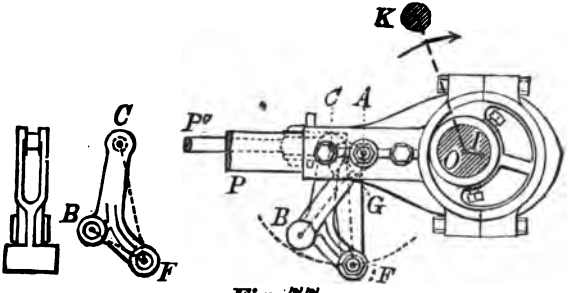

Fig. 77.

and the cut-off valve through a peculiar mechanism, consisting of four members, viz: the link, the rocker, the cut-off rod and the adjustment lever. The link *AB* being jointed by the pin *A* to the eccentric rod, imparts to the rocker a rocking motion on its fulcrum *F*. The rocker is of a peculiar shape, as shown detached in the cut, but it virtually represents a bell-crank (or angular lever), having an angle *BFC* of about 50°,

*Patented.

the arm CF of which is about twice as long as the arm BF. To the extreme end C of this rocker is jointed the cut-off rod, by which the cut-off valve is moved. For the purpose of changing the degree of expansion, the fulcrum F of the rocker can be moved in a circular arc, being attached to the end of the adjustment-lever GF, whose fulcrum G is a fixed point.

The study of this gear will consist in the investigation of the movement of the cut-off valve for several positions of the adjustment-lever. In every case we shall proceed from the neutral position of the rocker (found by transferring the eccentric rod to the centre of the crank-shaft) remembering that the movement of the mechanism will be symmetrical to both sides of this position. Besides, we shall as usual neglect all complicating influences resulting from the angularity of the several members; and besides, we shall assume the movement of the pin A to be strictly circular and coincident with the movement of the eccentric I.

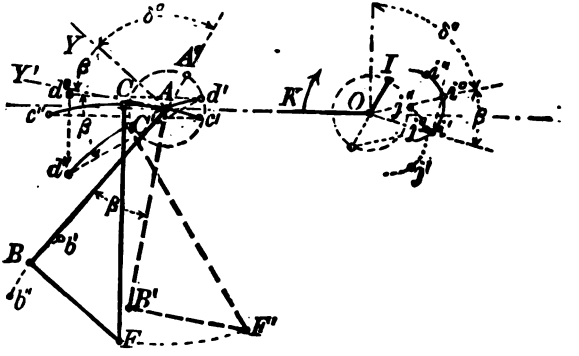

Fig. 78.

At first we move the adjustment-lever until the line CF of the rocker assumes a vertical position (see Fig. 78), the theory

for this position being the least complicated. When the mechanism is in operation the pin A will move in a circle, and hence the points B and C will move in the arcs $b'b''$ and $c'c''$. For the latter arc we shall substitute the chord to simplify the theory. When the crank is on its centre the pin A will occupy the position A° corresponding to the position of the eccentric I, and since the link AB represents, as it were, the eccentric rod for the cut-off gear, we can measure the angle of advance, $\delta^\circ = YAA^\circ$, by drawing AY at right angles to BA. The angle CFB being 50°, and AB being at right angles with FB—or approximately so—it is evident that the angle $YAA^\circ = \delta^\circ$ exceeds the angle of advance of the eccentric, I, by 50°. Owing to the dimensions of the rocker, the travel of the point C, and consequently also the travel of the cut-off valve, equals double the travel of the main valve; and hence we can find the ideal eccentric i° of the movement of the cut-off valve for the considered grade by advancing the line OI through an angle of 50° and doubling its length.

Next we move the fulcrum F towards the right to I' to change the grade, and denote the angular change of the rocker by the Greek letter β. The corresponding angular change of the link AB is practically the same, and the angular advance is consequently farther increased by this angle. We can therefore draw the line Oi', but we have yet to find its length. The movement $d'd''$ of the pin C' of the rocker is doubtless the same as it was before; but being inclined, it is only its horizontal component $d'd^\circ$ that is transmitted to the valve, and the throw Oi' of the ideal eccentric for this grade will be less than Oi°. The necessary reduction can be made by drawing the line $i^\circ i'$ at right angles to Oi', which will be understood when we consider the similarity of the triangles $Oi'i^\circ$ and $d'd^\circ d''$.

In moving the fulcrum F of the rocker in the opposite direction we would have found the ideal eccentric i'', and other

positions of the fulcrum F would furnish more points of the locus of ideal eccentrics The angle $i°i'O$ being a right one, it will easily be understood that all the ideal eccentrics will be located in a circle of which the line $Oi°$ is a diameter.

These results relate to the absolute movement of the valve and to find the ideal eccentrics for the relative movement, we move the locus in the direction of and through a distance equal to IO. Having thus determined the locus $j'j°j''$ of the relative movement, we can draw the valve diagram, Fig. 79, in the usual manner.

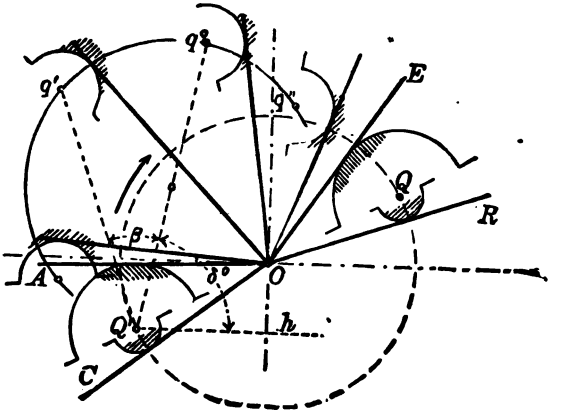

Fig. 79.

This diagram now shows that the cut-off can be adjusted to any point between the crank angles OA and OE, as the angular adjustment of the rocker is not limited. It shows, moreover, that this valve-gear is distinguished by the decided rapidity in cutting off. The closure of the steam passages is very sharp for all grades cutting off before the half-stroke; for a later cut-off, however, this rapidity will get less until, when cutting off at OE, the rapidity of the cutting off of the cut-off valve will about equal that o the main valve.

The rapidity of late cut-offs can be improved, if desired, by making the arm CF of the rocker more than double the length of the arm BF; whereby, the line Ci^o will be lengthened and consequently the locus-circle will be enlarged. This change entails an increase of the absolute movement of the cut-off valve above twice the travel of the main valve. Another measure consists in increasing the negative lap of the cut-off valve; but for reasons discussed in the foregoing chapter, it should never exceed the positive outside lap of the main valve. A reduction of the angle BFC of the rocker would likewise be efficient, but this reduction is attended by an increase of certain irregularities.

The sharpness of the cut-off will in reality be slightly less than indicated in the diagram, from the fact that the movement of the pin A is not circular as was assumed, but is more or less flattened.

The proportioning of the mechanism requires some care; for on it depends largely the proper operations and regularity of the cut-off. To this end we draw the rocker $B\ FC$ (Fig. 80)

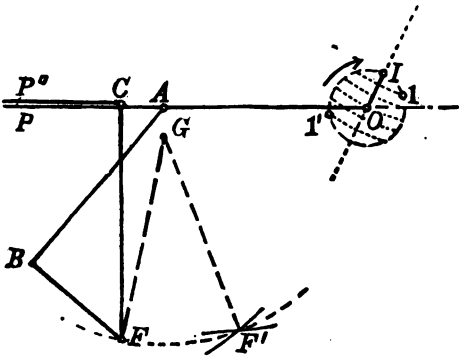

Fig. 80.

with the line CF in a vertical position, and the line BF at an angle of 50°, and make the arm BF from three to four times the throw of the eccentric, and the arm CF twice as long. (The given figures have been tested by a number of experiments). Then we draw the link AB at right angles to BF and make it about $\frac{2}{7}$ of the length of CF. The eccentric rod OP can next be shown in its neutral position, passing through the end A of the link. The cut-off rod $CP°$ can likewise be shown.

To find the length and position of the adjustment-lever GF, it is necessary to make a model of thin wood or veneer, consisting of the eccentric rod, the link, the rocker and the cut-off rod. Next we draw the orbit of the eccentric, and on it the exact position of the eccentric, say for every one-sixth of the stroke of the piston, which may be done in the following way, identifying the eccentric path with the path of the crank-pin: We divide the diameter of the orbit passing through the initial position of the eccentric 1 in six equal parts, and draw the projection arcs of the proper radius through those points, as shown. The next thing to be done is to attach the model to the drawing by joining the parts properly together with pins or thumb-tacks, and fastening the ends P and $P°$ of the rods to two slides representing the valves. The right end of the eccentric rod may be provided with a needle point which at first we put in the centre C, when we set the rocker directly over the position shown in the drawing, and mark the relative position of the two slides representing the valves. Then we make two additional marks on one of them, at a distance equal to the assumed negative lap of the cut-off valve, to show the relative position for the cutting off on either side. Suppose now we desire to find the proper position of the rocker for cutting off at the point 1. To this end we set the needle-point of the eccentric rod in the point 1 of the fore-stroke, and fix he valves for cutting off at the proper side, when we will find

that the end F of the rocker cannot be moved but in a certain curve. This curve we mark on the drawing by setting a needle-point into the rocker and making a slight scratch on the paper. Thereupon we attach the eccentric to the point $1'$ of the return stroke, fix the cut-off valve to the point of closure of the other passage of the main valve, and mark another curve by the point F of the rocker. The juncture F', of these curves, must of necessity, be the exact position of the fulcrum of the rocker when we desire to cut off at one-sixth of the stroke. In this way we can find the required position of the fulcrum for all the other grades, which will be found to form a curve. By substituting a circular arc for this curve, osculating as closely as possible, we obtain the location and length of the adjustment-lever GF. An arc can generally be found to agree with the constructed curve with an almost absolute precision; and hence it will be seen that this valve-gear will admit of a practically perfect equalization of the difference between fore and return stroke.

NOTES.

Steam Lap - or outside lap

Exhaust Lap - or inside lap

Throw = Eccentricity of the eccentric

Valve Travel

Equal to the double throw of the eccentric

Must be greater than the Exhaust lap on inside to open the exhaust port -

Must be greater than the Steam lap on inside to open the Steam port -

Steam port - Width equals the travel less the steam lap -

Exhaust port - width equals the travel less the exhaust lap -